PACKING AND DRIFT ELIMINATOR CHARACTERISTICS OF AN EVAPORTIVE CONDENSER

by

SUBROTO SEN

DEPARTMENT OF MECHANICAL ENGINEERING

INDIAN INSTITUTE OF TECHNOLOGY KANPUR

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CERTIFICATE

This is to certify that the work entitled '*Packing and Drift Eliminator Characteristics of an Evaporative Condenser*' has been carried out by Mr. Subroto Sen under my supervision and has not been submitted elsewhere for the award of a degree.

Keshav Kant 17.12.90

Dec, 1990

Dr. Keshav Kant
Assistant Professor,
Department of Mechanical Engineering,
Indian Institute of Technology,
Kanpur, India.

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NOMENCLATURE

a	Interfacial contact surface, m^2/m^3 of tower volume
C1	Cost of drift loss, paise per hour.
C2	Cost of power loss, paise per hour.
CADE	Cement asbestos drift eliminators.
CDE	Concrete drift eliminators.
COP	Coefficient of performance.
g	Acceleration due to gravity ms^{-2}.
G	Air loading, $kg_{dry-air}$ /h-m^2
h	Pressure drop in mm of water.
HDE	Honeycomb drift eliminator.
K	Overall mass transfer coefficient, kg/h-m^2 of contact area -(kg_{water} /$kg_{dry-air}$).
L	Water loading , kg_{water} /h-m^2 .
m_a	Mass flow rate of air , kg/min.
m_d	Specific drift loss , kg_{water}/$kg_{dry-air}$.
M_d	Rate of drift loss , kg/hr.
m_e	Specific evaporation loss, kg_{water}/$kg_{dry-air}$
n	Number of stages.
p	Static pressure at a point, mm of water.
P_d	Discharge pressure of compressor, bar.
P_s	Suction pressure of compressor, bar.
P_1	Power input to the compressor, kW.
P_2	Power input to the FD fan, kW.

P_3	Power input to the ID fan, kW.
t_1	Dry bulb temperature (DBT), C of the inlet air
t_2	Dry bulb temperature (DBT), C of the discharge air without the heater.
t_3	Dry bulb temperature (DBT), C with the heater on.
ϕ_1	Relative humidity (RH), of the inlet air (%)
ϕ_2	Relative humidity (RH), of the discharge air without the heater (%).
ϕ_3	Relative humidity (RH), of the discharge air with the heater on (%).
θ	Angle of inclination of the drift eliminators
T_1	Compressor outlet temperature, C.
T_2	Condenser outlet temperature, C.
T_3	Compressor outlet temperature, C.
T_4	Evaporator outlet temperature, C.
T_5	Temperature at the top of the packing, C.
T_6	Temperature below the packing, C.
T_7	Temperature of water in tank C.
v	Velocity of inlet air through inlet duct, m/s.
VAC	Supply voltage, volts.
V	Packing height, m^3/m^2
W1	Ambient air specific humidity, $kg_{water}/kg_{dry-air}$
W2	Specific humidity of the outlet air with the

	duct heater off, $kg_{water}/kg_{dry-air}$.
W3	Specific humidity of the outlet air with the duct heater on, $kg_{water}/kg_{dry-air}$.
	Density of water, kg/m^3.
P	Total pressure drop, mm of water.
	Inclination angle.
Z	Height of the packing, m.

Packing and Drift Eliminator Characteristics of an Evaporative Condenser.

ABSTRACT

The pressure drop and drift loss characteristics of thre different types of drift eliminators in an evaporative condense are experimentally investigated in both forced and induced flows These eliminators were Cement Asbestos Drift Eliminators (CADE) Concrete Drift Eliminators (CDE), and Honeycomb Drift Eliminator (HDE). For CADE and CDE data were taken for various number o stages, n and for a range of the orientation angle, θ of the eliminator plates. The COP of a refrigeration system was also studied with respect to θ. For HDE, the data were recorded for various values of n and the air velocity through the eliminator.

The results show that the pressure drop and drift are strong functions of n and fan speed. In case of CADE and CDE they are strongly dependent on θ as well. The characteristic curves for the three types of drift eliminators are reported.Based upon the cost analysis the breakeven values of θ for CADE and CDE lie between 50° and 60°. The COP of the refrigeration system decreases with increase in θ.

A new type of cellular packing placed inside the evaporative condenser was also studied experimentally. The volume transfer coefficient for the same is correlated in terms of water to air loading rates and the pressure drop per unit depth for it is also determined. The provision of a packing in the evaporative condenser results in COP values higher than those obtained without packing.

CHAPTER 1

INTRODUCTION

1.1 General Background

Evaporative cooling refers to two mechanisms namel simultaneous heat and mass transfer. These driving forces namel the temperature and the vapor potential play an important role i the cooling of water. The temperature of the water, which i sprayed into a flowing stream of air is reduced because of th evaporation of the water and the subsequent absorption of th latent heat by the air stream. Temperature reduction also takes place because of the sensible heat transfer from the water to the air stream.

This is the principle of cooling which finds widespread use in the cooling towers for power plants, air conditioning systems, in the chemical industries, evaporative condensers and also in desert coolers. In the case of evaporative condensers, the water which is sprayed over the condenser tubes, takes away some of the heat from the condenser coil due to sensible heat transfer while the air whose temperature is slightly raised by the evaporation of the water droplets also removes a portion of the heat. The water falling below the condenser is cooled by the process of adiabatic saturation.

In recent years, the demand for industrial water has increased

tremendously and at the same time the sources of water for industrial use are limited. For offsetting this, instead of once through cooling systems, cooling towers have been used. One of the problems with cooling towers was that it lets go a small percentage of the circulating water as drift, which apart from being a cost , is also hazardous environmentally.

To tackle this problem an efficient mechanical system is required to retain the water droplets flowing out along with the exit air stream. This loss of water droplets is known as 'drift'.

One method of solving this problem is to use some kind of drift eliminators (DE). These are installed above the spray system of the cooling tower or the evaporative condenser. They work on the inertial separation principle. Typically conventional DE consist of rectangular plates which are set at a desired angle to the air flow direction, over the whole outlet area of the air flow passage. By adjusting the angles of these plates in the flow direction suitably, most of the water droplets can be made to fall back into the sump, thereby reducing the drift loss.

There are some non conventional types of DE's which have been used in practice. These are the cellular type which are made from some honeycomb structures. They also work on the inertial separation principle. They can be molded out of low density plastics and therefore offer certain advantages in terms of their simplicity in manufacturing and their transportability.

For DE while on one side it is essential that the drift loss be reduced, but at the same time the pressure drop across the DE should not become high as this would lead to an additional cost of

power consumed by the fans. The design of DE, therefore should seek an optimum between these two opposing effects.

The effectiveness of a DE is an essential aspect coolingtower design for many reasons some of which are enumerated below:

1. Conservation of water.
2. Avoiding fan blade corrosion in case of induced draft fan.
3. Minimizing the chances of the bacteria 'Legionella' (causing the Legionnaire's disease) being carried along with the drift.
4. Avoidance of violation of local area environmental protection regulations.

For evaporative cooling, an essential requirement is the provision of an interfacial area between the water and air. The higher this area, the greater the cooling of water. Also greater the residence time of the water droplets in the tower, will lead to better heat and mass transfer .To provide these, i.e. greater contact area and higher residence time, some particular type of packing is used in cooling towers. Though different types of packings are used in the industry, very little published data are available on these.

The present study concerns itself with the performance characteristics of multistage cement asbestos, concrete and honeycomb drift eliminators with forced and induced flow separately.The effect on the performance of a refrigeration system is also studied. Finally, a new type of packing which has been designed and incorporated in the system is tested and evaluated for

its performance characteristics.

1.2 Scope of the Present Work.

The major objectives of this dissertation are to study the following:

1. The pressure drop and drift loss characteristics, for various orientations of drift eliminators made of concrete and cement asbestos in an evaporative condenser with a packing of cellular type and determine optimum angles of orientation for drift eliminator plates.
2. The pressure drop and drift loss characteristics for a cellular type of drift eliminator made of honeycomb structure.
3. Comparative study of the characteristics with forced and induced flow separately.
4. The performance characteristics of a new type of cellular packing.
5. The effect of the inclination angle of drift eliminators on the performance of a refrigeration system.

1.3 Organization of the Thesis.

Chapter two deals with the literature survey on the drift eliminators and the packings used in cooling towers. Chapter three gives details of of the test rig, the instrumentation

used and an outline of the experimental procedure. Also included are the details of the method for estimating the drift loss. Chapter four entitled 'Results and Discussion ' gives the data collected in the course of the experiments and the discussion of the results including the conclusions derived from this study. Suggestions for future work are also included here.

CHAPTER 2

LITERATURE SURVEY

2.1 Introduction

In most air conditioning and refrigeration systems and also in several industrial processes a certain amount of heat is generated which has to be rejected to the ambient. This is generally accomplished by making use of different types of heat exchangers. One such category of heat exchangers is the direct contact type which makes use of the principle of evaporative cooling as explained in section 1.1.

In the former i.e. the cooling tower the water to be cooled is pumped to the top of the tower and sprayed with the help of a spray distribution system. The ambient air is either forced or drawn through the tower. This air comes in direct contact with the water stream flowing down, resulting in the cooling of water by the evaporation of a small amount of it and also by sensible heat transfer. The air gets heated up and humidified and is discharged to the ambient. In order to increase the effectiveness of such a transfer process, generally the large towers are provided with a fill material or packing which can be of several designs. The basic function of the fill material is to increase the residence time of the two fluid streams.

The evaporative condenser on the other hand generally (see details in section 2.2) does not make use of a fill material. The coil is installed inside a cooling tower like configuration.The effectiveness of such an equipment depends upon the air and water loading rates, type of spray pattern and the wet bulb temperature of the inlet air.Although the introduction of a fill material in an evaporative condenser may improve its performance, but no published data are available in this regard. The characteristics of any new fill material will of course have to be studied.

The total surface area of the water droplets presented to the air stream directly depends on the spray distribution system and naturally affects the performance of the cooling equipment. The hot and humid air leaving the unit carries with it water droplets suspended in the air stream. This represents a direct loss of water may also be harmful to the environment as well as human beings. This loss can be prevented by an effective design of the drift eliminators. The best drift eliminators can be chosen or designed only after assessing the drift loss from a from a given configuration and for the given spray system.

2.2 Evaporative Condenser

The basic purpose of an evaporative condenser is to reject heat from a condensing vapor into the environment. It may be considered as a combination of a water cooled condenser and an air cooled condenser, wherein the heat rejection takes place by the evaporation of water into the air stream moving across a condenser coil.

Evaporative condensers eliminate the need of pumping and the chemical treatment of large quantities of water associated with cooling towers. They use much less fan horse power than air cooled condensers of the same capacity. In an air cooled condenser, heat rejection is limited by the ambient dry bulb temperature whereas in an evaporative condenser, it is limited by the ambient wet bulb dry temperature, which may be 6 to 15°C lower than the ambient dry bulb temperature. The most important aspect of the evaporative condenser is the fact that the systems using them can be designed for lower condensing temperatures. This results in lower compressor energy input when compared to the systems utilizing water or air cooled condensers.

In an evaporative condenser the vapor to be condensed is circulated through a condensing coil, which is continously wetted on the outer surface by a recirculating water spray system. Simultaneously air is made to flow over the coil causing a small amount of the recirculated water to evaporate. This evaporation of water results in the removal of heat from the coil thus cooling and condensing the vapor inside the coil. The main components of an evaporative condenser include the condensing coil, spray water pump, water spray system, induced or forced draft fan, drift eliminators and make up water system.

The high rate of energy transfer from the wetted surface of the condensing coil eliminates the need for fins. The water lost through evaporation, drift and blowdown is made up by the make up water assembly.

Most evaporative condensers use fans either to force the air

or to draw it through the unit. These fans may either be of the centrifugal or the propeller type. Drift eliminators are used to recover entrained moisture droplets from the discharge air stream. Most of the entrained moisture, is recovered, by the drift eliminators but some of it is discharged as drift anyway. The rate of drift loss from an evaporative condenser is dependent upon the unit configuration, eliminator type and design air and water flow rates. An effective drift eliminator design can reduce the drift loss to a range of 0.002 to 0.2 % of the water circulating rate (ASHRAE, 1983).

Heat transfer in an evaporative condenser takes place from the condensing vapor inside the tubes, through the tube wall to the water film outside the tubes and then from the water film to the air.

The rate of heat flow from the vapor to the water film is given by:

$$q = U_s A\,(t_c - t_e) \qquad (2.1)$$

where U_s = overall heat transfer coefficient, $W/m^2\,°C$

q = rate of heat flow, W

t_c = saturation temperature at the pressure of vapor entering the condenser, °C.

t_e = temperature of water film surface, °C

A = outside surface area of the condenser tubes, (m^2)

The rate of heat transfer from the water air interface to the air stream is given by:

$$q = U_c A\,(h_c - h_e) \qquad (2.2)$$

where U_c = overall heat transfer coefficient from the water air interface to the air stream, $W/m^2\ °C/(kJ/kg)$.

q = rate of heat flow,

h_c = enthalpy of saturated air, kJ/kg

h_e = enthalpy of air entering condenser, kJ/kg.

Equations 2.1 and 2.2 have three unknowns U_c, U_e and t_e (h_e), these equations cannot be solved directly and require an iterative procedure (ASHRAE, 1983).

2.3 Types of Drift Eliminators:

Drift eliminator refers to baffling that causes the exhaust hot air containing entrained water droplets to change direction a number of times. The droplets hit the eliminator surface and fall back into the tower. Drift eliminators used have been made of timber, metals, plastics, concrete, cement-asbestos etc.in many configurations.In most old cooling towers, wooden slats placed at an angle to the direction of the air stream were used as drift eliminators. Presently also, most of the drift eliminators used are rectangular slats kept at an angle. Various configurations have been tried, but any increase in the effective water retention has always led to greater pressure losses.

Today the modern drift eliminators fall into two main categories ,'C' or 'Z' shaped blades which are plastic refinements of the wooden slat concept and the cellular type of drift eliminators which are extensions of the film fill concept. Irrespective of the shape, all drift eliminators work on the

inertial separation principle. The air is forced to change direction, but the water droplets being carried along with the air are unable to make the change as fast and as a result impact on the side of the eliminator blades. Once the small droplets have contacted a solid surface they coalesce to form a film on it and eventually drain from the drift eliminators as large droplets. The pressure drop across these blades can vary considerably from product to product, so each design should be evaluated carefully.

Studies have been made on cellular drift eliminators made out of neoprene asbestos by Burger et al. (1975). A cellular drift installation adds to the efficiency of the operation by stopping the air from making heavy turns and also collects any water droplets being carried out by the outgoing air. An additional bonus of using neoprene asbestos is that it provides a fire proof barrier.In a recent study, experiments were carried out on one, two and three stages of wooden drift eliminators, Das(1988) with forced flow. In another study of Singh(1988), the experiments were carried out with single, and double stage concrete drift eliminators, to determine an optimum angle of orientation for various operating conditions.

Chan and Golay (1977), used three configurations of drift eliminators in their experiments. These shapes were (a) Sinus shaped, (b) three straight segments and (c) a zig zag eliminator. The result of their experiments showed that as the complexity of the geometry increased, the pressure loss increased, but the drift loss decreased. This showed that it was necessary to strike a compromise in designing an eliminator to minimize the pressure loss

and increase the amount of water retained in the system (ie. reduce the drift loss).

2.4 Methods of Drift Measurement:

The drift from an evaporative condenser or a cooling tower refers to droplets that escape from it with the exhaust air and it is generally expressed as a percentage of the total water circulation rate It is a potential hazard both for the environment as well as for human beings. In order to minimize the drift emissions, cooling towers commonly make use of drift eliminators of some or the other type. Complete capture of the entrained droplets in the exit stream is not possible (Chan and Golay,1977), but to be able to decide the most effective type of drift eliminator, it is necessary to measure the emmissions from a tower.

Many different drift measurement devices and techniques are currently in use (Roffman, 1973; Rogener,1979; Policastro and Wastag, 1981) which differ widely in design, sophistication, underlying principles and capabilities. No single method can be considered as the best drift measurement technique. In the absence of a comparative analysis of various drift measurement devices, a comprehensive study on drift measurement methods was undertaken at M.I.T. by Golay et al. (1986), and many types of devices using different techniques and underlying principles were used for the purpose. A spectrum of test environments were created which spanned the range of cases normally encountered in cooling towers. The environment differed according the droplet mass flux, the droplet size distribution and gas speed.Cases tested included both

mechanical and natural draft cooling tower environments. A brief description of the different methods is given below :

2.4.1 Total Droplets Mass Flux Methods : These methods do not require the measurement of droplet sizes.

1. Cyclone Separator: The cyclone separator is designed to drift content of the exit stream. It extracts a continous isokinetic sample from the cooling tower exit stream. Centrifugal force is used to separate the drift droplets, causing the droplets to collide on the separator walls and then flow downwards as a film to a collection jar at the bottom of the cyclone. The isokinetic sampling system ensures that the drift droplets entering the system are representative of those entrained in the exit stream at the measuring point.
2. Heating Psychrometer: It is a thermodynamic device consisting of an aluminium pipe with heating coils inside its inlet section and a psychrometer. The psychrometer is placed downstream of the heater. The initial psychrometric data of the sample are noted after which the heater is switched on. The drift droplets evaporate and the new psychrometric values are noted. The drift or the droplet content of the gas stream is given by the difference between the humidity ratios of the two states.
3. Moisture Condensation Device: In the operation of this device, a gas sample is cooled inside a cooling helix. The condensing moisture produces a film in which the entrained water drift droplets as well as its residue are trapped. The

liquid flowing down the cooling helix is collected in a container. The electrical conductivity or concentration of a chemical tracer is then measured and the drift mass flux is inferred from the principle of conservation of salt or tracer material The only assumption made here is that the concentration in the entrained droplets is the same as in the recirculating water flow.

2. Droplet Size Dependent Number Flux Methods

1. Coated Slide Technique: In this technique a droplet impaction method is used. The droplets impact on carbon black coated glass slides presented perpendicular to the direction of the exit air stream. A droplet impinging on the glass slide would produce a tiny crater from which the droplet size can be determined.

2. Sensitive Paper Method: In this method the air stream is forced to change the direction of flow. The obstructions causing the change of direction are called collection planes and are covered with sensitive paper. The droplets impact on this paper by inertial separation and cause a chemical reaction in this paper. This chemical reaction reaction leaves a droplet size dependent stain which can be measured.

3. Droplet Size Dependent Number Density Methods.

1. Laser Light Scattering Technique: In this method a pulsed laser light scattering device is used. The intensity of the pulsed diode laser light , scattered by the droplets is measured in the form of a voltage pulse. It is assumed that the height of the voltage pulse caused by the scattered light

is proportional to the square of the droplet radius. This pulsed system measures only the instantaneous number and mass densities typical of a drift population.

2. Droplet Photography: The system uses a special camera lens having a well defined narrow depth of field. Photographs of droplets are taken within a sensitive region located between the strobe and the camera. The developed negatives are analyzed by an automatic photographic data analyzer to get the required results.

The study concluded that the mutual agreement of the results obtained with the different techniques examining the same situation was generally poor. Many devices perturb the flow mechanically due to their physical presence or due to impaction. These effects contribute substantially to the measurement error for most such devices and are either very difficult to or almost impossible to eliminate.The final conclusion of the study was that no single method enjoyed a universal acceptance as the best drift measurement technique. Droplet sizing methods were more effective in low load, small droplet size spectrum situations. Isokinetic mass and chemical assay techniques (drift mass flux methods) are most accurate in high load large droplet situations. Measurements relying upon thermodynamic measurements were reliable within an order of magnitude only.

2.5 Spray Water Systems:

The function of the spray system is to ensure a uniform distribution of water of the condenser tubes. If it is not uniform, some areas may be flooded and some may remain dry. The latter are the regions of underperformance.

Two types of spray systems are used namely the gravity and the spray type. The gravity distribution system if employed is for cross flow towers. The water reservoir is left open to the atmosphere and the water gravitates through orifices to the packing below. The spray type is usually employed for counterflow towers. The water is sprayed through nozzles under a certain head.

Presently for cross flow towers, the best water distribution system incorporates 'target nozzles'. These consist of an adjustable and,or removable orifice and a plate or a targetthat is suspended about 100 mm below the orifice. The target further breaks up the spray and spreads the water jet over the fill. These nozzles are placed fairly close to each other and a certain amount of overlap of the spray patterns ensures adequate coverage. For counterflow towers of small and medium capacities,use is made of medium pressure nozzles (0.2 bar to 1.1 bar). These provide solid cone patterns with little or no overlap. THe best systems of this type use the square pattern nozzle. Burger(1977)found the square pattern of nozzles to be very efficient

For large towers used in power plants, the nozzles are of the pressure(< 0.3 bar), and have hollow cone round patterns A large amount of overlap is provided which have been shown to result in the most even water distribution possible.

In a study carried out by Howard et al (1941), on different types of nozzles, it was found that the performance was affected more by the drop size distribution than by the water distribution within the tower.

2.6 Types of Packings

The packing in a cooling tower plays a very important role as far as the effectiveness of the tower is concerned. It is in this region that the heat and mass transfer and the related cooling of the water takes place.

The packings used are mostly of two types, the splash fill type and and the film fill type. Splash fill consists of a number of slats of various designs kept horizontally inside the cooling tower. The water spray splashes over these slats.The presence of a fill in a cooling tower increases the residence time of the water compared to the situation of no fill in the cooling tower.

The second and the most common type of packing used is the film type packing which generates films of water. The big advantage of this, compared to the splash type of packing is that it slows down the water passing through the tower to a greater extent.This allows for a much shorter packing section. Another advantage is that the heat transfer surfaces are parallel to the air flow rather than perpendicular to it as it is in the case of splash bars. Kelly et al (1956), investigated eight different types of packing arrangements of the splash grid type. The shapes used by him were rectangular slats kept horizontally or at an angle to the horizontal. Also square cross-section slats, kept with

their diagonals horizontal were used. The size of these slats and the distance between them was varied.

Another type of packing used by Narayankhedkar et al (1977), consisted of alternately arranged flat and corrugated aluminium strips. Lowe and christie (1961), used corrugated asbestos cement louvers in both splash and film arrangements. Many authors in the past have used the standard splash type wooden slats like Lichtenstien(1943), Uchida et al(1966), Huchinson(1942), Sherwood et al(1946). Honeycomb board was used by Uchida et al (1966), Simpson and Sherwood (1946),used masonite sheets.

From the above it is clear that plenty of work is going on, on different types packings. The aim of which is to find packings having good performance characteristics.

2.7 Packing correlations

During the past few years,there has been a large increase in the use of evaporative condensers and cooling towers. Despite this there is very little published information on the performance characteristics of different types of packings.

In an investigation on splash grid type of wooden slat packing (Kelly, 1961), it was found that its transfer characteristics could be related by the equation:

$$KaV/L = 0.07 + AN(L/G)^{-n} \qquad (2.3)$$

where A, n = constants for a particular type of packing,

N = number of decks.

The constant 0.07 appears because of the end effects in the sections above and below the packing and also below the spray

system.

In another investigation on corrugated aluminium rolls used as packing (Narayankhedkar,1977), the characteristic for the packing was found to be:

$$Ka = C (L)^{0.613}(G)^{0.805} \tag{2.4}$$

Lowe and Christie(1961) in another experiment found the correlation of the type:

$$Ka/L = \lambda (L/G)^{-n} \tag{2.5}$$

where λ and n are constants for the type of packing. The cooling tower Institute in Houston (Texas) recommends the following relationship of the type, packings

$$KaV/L = \lambda (L/G)^{-0.6}$$

For film fill packings made of asbestos sheets the relationships found by Lowe et al(1961), were: (2.6)

$$Ka/L = 0.185(L/G)^{-0.62} \tag{2.7}$$

$$Ka/L = 0.097(L/G)^{-0.66} \tag{2.8}$$

These relationships were for two different settings of the corrugated asbestos sheets. For splash type of packings the relationships for two settings of the splash bars were:

$$Ka/L = 0.094(L/G)^{-0.5} \tag{2.9}$$

$$Ka/L = 0.145(L/G)^{-0.71} \tag{3.0}$$

CHAPTER 3

EXPERIMENTAL RIG AND PROCEDURE

3.1 Test Rig

The complete test rig consisted of an evaporative condenser with a honeycomb packing. In order to study the effect of both forced and induced flow, use was made of an induced draft(ID) and a forced draft(FD) fan. The drift eliminators used were of three kinds; namely concrete, cement-asbestos and honeycomb. The evaporative condenser installed was one of the components of a vapor compression system using R-22 as the refrigerant. The main components of the test rig are shown in fig. 3.1. Their description and specification are as follows:

1. Refrigeration system consisted of the following:

   a. Compressor:

      Make: Shriram, 1.5 ton capacity, R-22 compressor

      Voltage: 220 VAC; Amps: 12.2; RPM: 2850

      Evaporating Temperature: 7.2 °C;

      Condensing Temperature : 55 °C

      Ambient Temperature : 35 °C

      Compressor suction gas Temperature: 35 °C

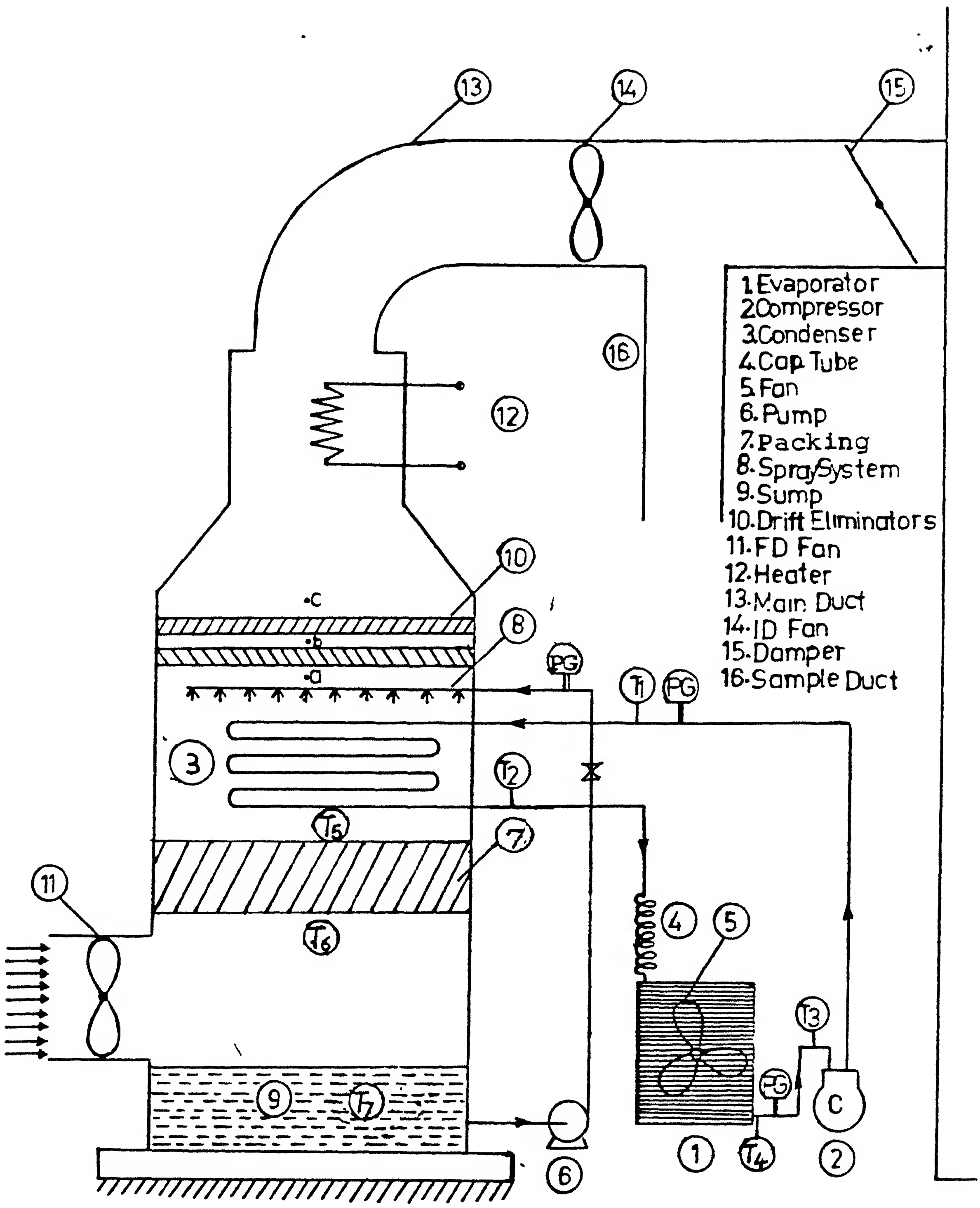


Fig 3.1
Schematic Diagram of the Test Rig

Suction Pressure : 5.25 bar

Discharge Pressure : 21.467 bar

b. Condenser:

Designed for a capacity of 1.5 tons. It was made of 3/8 inches (9 mm) copper tubes of length 23.1 m.

c. Capillary tube:

It was provided for a rated capacity of 1.5 tons as an expansion device having a total length of 0.82 m.

d. Evaporator:

The evaporator used was a 1.5 ton finned air-cooled unit. A fan was installed a the back of this unit to blow ambient air for better heat transfer to the refrigerant.

2. Main chamber consisted of:

a. A rectangular box 1.00 m X 0.90 m X 1.40 m.

b. A DE chamber 1.00 m X 0.52 m X 0.52 m having a tapered top portion and,

c. a heater box 0.45 m X 0.45 m X 0.45 m .

The main chamber was made of slotted angle iron frame and aluminum sheets. It housed the spray system ,condenser coil and the packing. The DE chamber housed eliminators supported on angles. The heater box housed finned duct heaters of a total capacity of 6kw on top of the DE chamber. The bottom portion of the main chamber was used as a water sump.

3. Drift Eliminators: For the present experimentation three types of eliminators were used. The box in which they were installed was of

the size (900 mm X 500 mm X 50 mm).

a. Cement Asbestos Drift eliminators (CA DE):
Size: 895 mm X 50 mm X 6 mm) . Clearance at 90° = 10 mm
Number of strips in a stage: 8 .

b. Concrete Drift Eliminators (CDE):
Size : 895 mm X 50 mm X 25 mm. Clearance at 90° = 13 mm
Number of strips in a stage: 7.

c. Honeycomb Drift eliminator (HDE) : These were of a somewhat complicated shape. Each strip consisted of circles of 25 mm diameter and were 15 mm in height, which were laid side by side. Three such strips made up a single DE. The strips were arranged in a staggered having a overall size of 900 mm X 450 mm X 45 mm.

4. Circulating water pump: It was installed to provide the spray water over the condenser coil.
Pump Specifications: Power 2 HP, 3 Phase, voltage 415 V, RPM 2850, Amp 3.8, Make Beacon Monobloc, Type IDM5 - 143.

5. ID and FD Fans: FD fan and motor assembly was installed on a concrete foundation by six foundation bolts. ID fan was mounted on a specially raised platform
Specifications: Fan discharge 89.9 m /min, Static pressure 13.5 mm of water, diameter 250 mm.
Motor: 0.75 HP, 1420 RPM, 5.2 Amp, 230 V, single phase.
Make Premier Corporation (India) Ltd., Coimbatore.
ID Fan specification are the same as those of the FD Fan

specifications

6. Packing: The packing used was of a new type. It was made up of seven strips kept one over the other in a staggered fashion. Each strip consisted of a number of circles of diameter 25 mm and 15 mm height, laid side by side. The overall dimensions of the packing were 1.05 m X0.89 m X 0.51 m. The area of the packing was 0.453 m.
7. Ducts: A main duct was connected from the top of the heater box to the inlet of the ID Fan which had been mounted on a raised platform. The main duct was made up of two sections consisting of a 90° duct of cross - sectional area 460 mm X 280 mm and a connecting duct of rectangular cross-section 460 mm X 280 mm at one end and a circular section of diameter 360 mm at the other end, having a total length of 1.3 mm. The discharge duct was connected to the outlet of the ID Fan to carry the discharge out of the room. The dimensions of this duct were 2.26 m X 0.41 m X 0.32 m and was equipped with a damper which when closed permitted the flow of the air in the sampling duct. The dimensions of the sampling duct were 1.20 m X 0.18 m X 0.18 m. It was used to measure the psychrometric conditions of the air at its outlet.

3.2 Instrumentation

Following instruments were used to measure the pressure, temperature, relative humidity and the air velocity at various points:

a. Pressure gauges

i. Compressor suction side pressure gauge

Range: -2.07 bar to 10.34 bar (gauge).

ii. Compressor discharge side pressure gauge:

Range: 0.00 bar to 20.68 bar (gauge).

iii. Water pump discharge side pressure gauge:

Range: 0.00 bar to 3.45 bar (gauge).

b. Temperature sensors: Copper-Constanton thermocouples.

c. Temperature recorder:

Honeywell Temperature Recorder Electronic - 15.

Range: -40 F to 300 F .

d. A vacuum pump was used to evacuate the refrigeration system before charging the refrigerant.

e. A water tank of known volume capacity and a stopwatch were used to measure the flow rate of the water sprayed on the condenser coil and to calibrate the water pump pressure gauge.

f. The psychrometric data of the inlet and outlet air was measured using a relative humidity sensor which gave the relative humidity and the dry bulb temperature (DBT).

Range: -8 to 76 °C , 0 to 100 % RH . Make: Cole Parmer

Model No. : 3310 - 40 .

g. Velocity of air was measured using a vane anemometer at the inlet duct to the FD fan.

Range : 0 - 10 m^{σ}, Wind speed :1 - 15 m/s .

Make : Ota Keiki Seisakusho, Japan.

h. Power input to the compressor, FD and ID fan were measured by using watt meters.

Watt meter (for the compressor):

Make: Nippen, Class: 1.5 I.S. 1248-68.

Model: SF-144P-IEW, Amp: 20A.

Voltage : 250 V, Range: 0 - 5000W.

Watt meter (for the FD and ID fans):

Make: Toshniwal. Range: 0 - 2.5 kW.

Least count : 0.5 kW.

i. For varying the voltage input to the FD and ID fans, a variac was used.

Range: 0 - 270 V , Max. load : 15 Amps.

Make: Automatic Electric Pvt. Ltd. , Bombay.

j. The pressure drop across the drift eliminators was measured using an inclined tube manometer.

Range : 0.1 to 1.0 in. ,Make: Dwyer.

Specific gravity of oil: 0.826.

3.3 Experimental Procedure

The steps of the experimental procedure are as follows:

1. Before starting the experimental run all power connections and water connections were checked to ensure safety of the setup.

2. Cement asbestos drift eliminators were set at an angle of 90 to the horizontal and were inserted into the drift eliminator box.
3. The FD fan was run at 230 VAC.
4. The pump was switched on to provide water to the nozzle spray system, and the pressure controlled in order to fix the flow rate of water.
5. The refrigeration cycle was switched on.
6. After about twenty minutes or so when the system had stabilized, the suction and discharge pressures of the compressor temperatures at various points of the refrigeration cycle and those of the evaporative condenser were measured.

The pressure drop across the drift eliminators, the power consumed by the FD fan and the compressor were recorded. The dry bulb temperature (DBT)and the relative humidity (RH)were recorded for the entering air at the inlet to the main chamber and at the outlet of the sample duct. The velocity was also measured at the inlet to the main chamber.

7. The duct heaters mounted on the top of the drift eliminator box were switched for the evaporation of the water droplets carried by the outgoing air.
8. After about twenty minutes, readings for the DBT and the RH were repeated and the heaters were switched off.
9. Steps 4 - 8 were repeated for the FD fan with supply voltage 200 and 160 VAC.
10. Above set of readings were repeated for inclination angles of 75°, 60°, 45°, 30° and 15°.

11 Steps 2 to 10 were repeated for two stage and three stage cement

asbestos, three stage concrete drift eliminators with FD fan.

12 Steps 2 to 9 were repeated for one, two and three stages of honeycomb drift eliminators.

13. The ID fan was then used instead of the FD fan and steps 1 to 12 were repeated.

13. Make up water rate was also measured.

In order to determine the performance characteristics of the packing , following procedure was used:

1. All power and water connections were checked.
2. Three stages of honeycomb drift eliminators (HDE) were inserted into the drift eliminator box.
3. The FD fan was run at 230 VAC.
4. The pump was started with a pressure setting of the pressure gauge at 1.35 bar (gauge).
5. The refrigeration cycle was switched on.
6. After about twenty minutes, the temperatures at various points of the refrigeration cycle, the temperature of the water in the water sump, the temperatures just above and below the packing were noted. The DBT and RH of the air were noted both at the inlet of the main chamber and at the outlet of the sampling duct. The inlet air velocity was also measured.
7. Steps 1 to 6 were repeated for different settings of the pressure discharge of the water pump, i.e. at different flow rates.
8. Steps 1 to 7 were repeated with the ID fan instead of the FD fan.
9 The steps steps 1 to 6 were repeated for various settings of the voltage supplied to the FD fan and the pressure drop across the

packing was measured.

10. Steps 1 to 6 and step 9 were repeated for 2 stage and single stage HDE.

3.4 Estimation of the Drift Loss.

Psychrometric data were measured for the entering air and the air leaving the sampling duct with and without the duct heaters switched on.

When the heater is not switched on, a simple mass balance of the dry air and the moisture entering and leaving the main chamber is given by:

$$m_{a1} = m_{a2} = m_a \qquad - \quad 1$$

$$m_{a1} W_1 + m_e + m_d = m_{a2} W_2 + m_d \qquad - \quad 2$$

i.e. $$m_e = m_a (W_2 - W_1) \qquad - \quad 3$$

Now when the heater is switched on , the drift going through the drift eliminator gets evaporated. This will alter the quality of the outgoing air. Mass balance yields:

$$m_{a1} W_1 + m_e + m_d = m_a W_3 \qquad - \quad 4$$

or $$m_e + m_d = m_a (W_3 - W_1) \qquad - \quad 5$$

substituting m_e from equation 3 in 5 we get

$$m_d = m_a (W_3 - W_1 + W_1 - W_2).$$

i.e. $$m_d = m_a (W_3 - W_2) \qquad - \quad 6$$

where m_{a1} , m_{a2} = mass of dry air entering and leaving the evaporative condenser. m_d = rate of drift loss $kg/kg_{dry\ air}$

m_e = rate of evaporation loss $kg/kg_{dry\ air}$.

CHAPTER 4

RESULTS AND DISCUSSION

4.1 Introduction

In order to carry out the objectives of this dissertation as outlined in section 1.2, experiments were conducted as per the procedure given in section 3.3. The psychrometric data were taken with drift eliminators made of cement asbestos, concrete and also those of the low density polypropylene in the honeycomb configuration. Pressure drop and temperature were taken for the cellular packing made in the honeycomb configuration. As the evaporative condenser used in the set up was hooked up to a 1.5 ton vapor compression refrigeration system, necessary data were also recorded for the refrigeration system to estimate its performance.

4.2 Drift Loss

The drift loss was calculated as a percentage of the circulating water flow rate. The psychrometric data for determining the drift are given in Tables 4.1 through 4.6 for CADE and Tables 4.7 & 4.8 for CDE. The data were recorded for the angle of inclination of the drift eliminator plates being varied from 15° to 90°. For each angle of orientation the fan speed and in turn the air flow rate was varied by varying the supply voltage. The values used were 230, 200 and 160 VAC. The number of stages used

TABLE 4.1

Psychrometric and Pressure drop data for CADE
Angle of Orientation : 15°, Air inlet area: 0.1006 m^2
Make up water: 0.518 kg/min, Water flow rate: 80 kg/min

V (Volts)	No. of stages	Compressor power	Fan Power	Inlet conditions		Discharge air without heater		Discharge air with heater		Velocity	Pressure drop
	n	(kW)	(kW)	dbt t_1	ϕ_1	dbt t_2	ϕ_2	dbt t_3	ϕ_3	v	Δp
			(°C)	(°C)	(%)	(°C)	(%)	(°C)	(%)	(m/s)	(mm)
FD FAN											
230	1	2.25	0.65	26.39	85.0	26.35	100.0	30.83	92.5	11.47	0.00
200		2.20	0.60	26.67	88.0	27.22	100.0	31.11	93.0	10.80	0.00
160		2.10	0.60	26.67	90.0	26.67	100.0	30.83	99.5	10.00	0.00
230	2	2.22	0.60	26.94	88.0	28.60	100.0	32.00	97.0	11.33	0.13
200		2.25	0.59	27.78	87.0	28.60	100.0	31.67	96.5	10.67	0.00
160		2.25	0.72	26.94	86.5	28.33	100.0	32.22	90.0	9.80	0.00
230	3	2.10	0.57	26.39	85.0	27.22	99.5	30.00	97.0	10.87	0.25
200		2.25	0.52	26.39	85.0	26.39	100.0	31.94	86.0	10.53	0.13
160		2.25	0.51	27.50	89.0	28.33	100.0	29.44	99.0	10.00	0.00
ID FAN											
230	1	2.10	0.46	25.00	90.5	26.67	100.0	29.00	97.0	6.73	0.25
200		2.20	0.42	25.00	90.5	26.67	100.0	29.00	98.0	6.40	0.00
160		2.20	0.38	25.00	90.0	26.39	100.0	28.33	98.5	6.27	0.00
230	2	2.12	0.47	26.67	88.0	27.22	100.0	29.44	95.0	7.13	0.25
200		2.20	0.41	27.78	85.0	28.33	100.0	30.83	95.5	6.53	0.13
160		2.18	0.39	28.61	84.5	28.33	100.0	30.83	95.0	6.33	0.00
230	3	2.10	0.45	27.22	81.0	27.78	100.0	29.44	94.0	6.53	0.25
200		2.25	0.37	27.22	81.0	28.33	100.0	30.50	95.0	6.33	0.13
160		2.20	0.40	27.22	81.0	27.50	99.5	29.44	93.0	6.20	0.00

TABLE 4.2

Psychrometric and Pressure drop data for CADE
Angle of Orientation: 30°, Air inlet area: 0.1006 m^2
Make up water: 0.518 kg/min, Water flow rate: 80 kg/min

V (Volts)	No. of stages n	Compressor power (kW)	Fan power (kW) (°C)	Inlet conditions dbt t_1 (°C)	Inlet conditions ϕ_1 (%)	Discharge air without heater dbt t_2 (°C)	Discharge air without heater ϕ_2 (%)	Discharge air with heater dbt t_3 (°C)	Discharge air with heater ϕ_3 (%)	Velocity v (m/s)	Pressure drop Δp (mm)
FD FAN											
230		2.00	0.52	26.67	88.0	27.22	100	30.83	88.5	11.20	0.25
200	1	2.10	0.50	26.39	90.0	27.78	100	30.00	93.0	10.33	0.13
160		2.10	0.46	26.39	90.0	26.94	100	30.00	90.0	10.00	0.13
230		2.00	0.52	28.33	73.0	26.94	100	31.67	83.0	11.07	0.64
200	2	2.10	0.50	28.33	80.0	26.94	100	32.00	74.0	10.53	0.38
160		2.10	0.47	28.33	86.0	26.94	100	31.67	83.0	10.13	0.51
230		2.15	0.52	28.33	73.0	26.94	100	29.50	97.0	11.33	0.76
200	3	2.15	0.48	25.56	78.0	27.22	100	29.44	92.5	10.27	0.51
160		2.20	0.46	25.56	84.0	26.94	100	29.44	92.5	10.13	0.25
ID FAN											
230		2.10	0.45	26.67	82.0	26.94	100	28.33	96.0	7.00	0.25
200	1	2.20	0.40	26.67	82.0	26.67	100	29.44	96.5	6.87	0.25
160		2.20	0.36	26.67	92.0	26.67	100	28.33	97.0	6.60	0.13
230		2.25	0.45	26.11	90.0	26.67	100	29.17	95.5	6.80	0.38
200	2	2.30	0.40	25.00	91.0	26.94	100	28.89	95.5	6.40	0.13
160		2.32	0.38	24.44	94.0	26.67	100	28.89	94.5	6.27	0.13
230		2.05	0.43	26.94	80.0	27.22	95	30.00	90.5	6.67	0.38
200	3	2.20	0.39	27.22	81.0	28.33	97	30.56	92.0	6.07	0.13
160		2.20	0.37	27.78	80.5	27.78	98	30.83	86.0	5.87	0.13

TABLE 4.3

Psychrometric and Pressure drop data for CADE
Angle of Orientation : 45°, Air inlet area: 0.1006 m^2
Make up water: 0.518 kg/min, Water flow rate: 30 kg/min.

V (Volts)	No. of stages n	Compressor power (kW)	Fan power (kW)	Inlet conditions dbt t_1 (°C)	ϕ_1 (%)	Discharge air without heater dbt t_2 (°C)	ϕ_2 (%)	Discharge air with heater dbt t_3 (°C)	ϕ_3 (%)	Velocity v (m/s)	Pressure drop Δp (mm)
FD FAN											
230		2.40	0.55	26.67	90.0	27.22	100.0	30.00	96.0	11.40	0.51
200	1	2.40	0.52	26.67	92.0	27.22	100.0	30.00	98.0	10.73	0.76
160		2.35	0.49	26.67	94.0	27.22	100.0	29.72	94.0	10.53	0.25
230		2.20	0.51	26.67	92.5	28.06	100.0	31.67	96.0	11.73	0.56
200	2	2.20	0.48	26.67	92.5	28.00	100.0	31.67	94.0	11.20	0.51
160		2.20	0.47	26.67	92.5	28.06	100.0	31.67	92.0	10.33	0.33
230		2.30	0.57	26.94	84.5	28.39	99.5	32.33	87.0	11.67	0.76
200	3	2.35	0.50	26.67	92.0	27.78	100.0	31.94	87.0	10.93	0.51
160		2.40	0.49	26.67	92.0	27.78	100.0	31.94	91.0	10.80	0.46
ID FAN											
230		2.10	0.43	26.67	81.0	26.94	100.0	28.89	95.5	6.93	0.12
200	1	2.20	0.40	26.67	78.0	27.22	100.0	29.44	94.5	6.53	0.12
160		2.20	0.35	26.67	78.0	27.22	100.0	28.89	95.5	6.47	0.00
230		2.30	0.45	28.33	79.0	28.33	95.0	32.33	83.5	7.00	0.38
200	2	2.30	0.39	28.33	72.0	28.89	95.5	32.22	83.0	6.47	0.25
160		2.35	0.38	26.67	32.0	28.33	96.5	31.67	85.5	6.40	0.13
230		2.50	0.47	26.94	84.5	28.39	99.5	32.22	88.0	6.80	0.25
200	3	2.40	0.40	28.06	80.0	29.44	98.0	32.73	85.0	6.53	0.38
160		2.30	0.37	28.82	76.0	28.33	95.0	32.22	78.0	6.47	0.25

TABLE 4.4

Psychrometric and Pressure drop data for CADE

Angle of Orientation: 60°, Air inlet area: 0.1096 m^2
Make up water: 0.518 kg/min., Water flow rate: 80 kg/min.

V (Volts)	No. of stages n	Compressor power (kW)	Fan power (kW)	Inlet conditions dbt t_1 (°C)	Inlet conditions ϕ_1 (%)	Discharge air without heater dbt t_2 (°C)	Discharge air without heater ϕ_2 (%)	Discharge air with heater dbt t_3 (°C)	Discharge air with heater ϕ_3 (%)	Velocity v (m/s)	Pressure drop Δp (mm)
FD FAN											
230		2.40	0.55	26.67	89.5	27.22	100.0	31.11	95.5	11.33	1.02
200	1	2.50	0.52	27.78	87.0	28.89	100.0	31.35	88.5	10.53	0.95
160		2.40	0.47	27.78	87.0	25.33	100.0	31.67	91.0	10.13	0.64
230		2.50	0.57	27.22	82.0	28.33	100.0	32.22	88.0	11.53	1.27
200	2	2.50	0.50	28.06	82.0	28.33	100.0	32.22	86.0	11.07	0.97
160		2.50	0.47	28.06	83.0	28.89	100.0	32.22	85.0	10.33	[illegible]
230		2.45	0.57	28.33	86.0	28.33	100.0	32.22	88.0	11.67	2.27
200	3	2.40	0.50	28.39	80.0	27.78	100.0	32.50	82.0	11.40	2.16
160		2.43	0.47	28.39	81.5	28.35	100.0	32.78	83.0	11.00	1.98
ID FAN											
230		2.20	0.45	26.67	90.0	27.50	100.0	31.39	90.0	6.20	0.51
200	1	2.30	0.42	26.67	90.0	28.00	100.0	31.39	89.5	5.73	0.38
160		2.28	0.40	26.67	90.0	27.90	100.0	31.11	90.5	5.73	0.38
230		2.20	0.47	26.38	88.0	27.22	99.0	30.83	88.0	6.20	0.76
200	2	2.22	0.42	26.38	89.5	27.5	99.5	30.83	87.5	6.00	0.64
160		2.35	0.38	26.38	86.0	26.94	99.0	31.11	86.0	5.67	0.51
230		2.20	0.46	26.94	84.5	28.00	98.0	32.22	91.0	5.87	1.40
200	3	2.15	0.41	26.66	87.5	29.26	98.0	32.50	76.0	5.47	1.27
160		2.10	0.37	26.66	87.5	28.00	98.5	32.22	78.0	5.33	1.27

TABLE 4.5

Psychrometric and Pressure drop data for CADE

Angle of Orientation: 75°, Air inlet area: 0.1096 m^2

Make up water: 0.518 kg/min., Water flow rate: 80 kg/min.

V (Volts)	No. of stages n	Compressor power (kW)	Fan power (kW)	Inlet conditions dbt t_1 (°C)	Inlet conditions ϕ_1 (%)	Discharge air without heater dbt t_2 (°C)	Discharge air without heater ϕ_2 (%)	Discharge air with heater dbt t_3 (°C)	Discharge air with heater ϕ_3 (%)	Velocity v (m/s)	Pressure drop Δp (mm)
FD FAN											
230		1.95	0.60	26.67	84.0	26.94	100.0	29.44	94.0	10.87	1.52
200	1	2.00	0.56	26.39	85.0	26.67	100.0	30.00	[illegible]	10.67	1.52
160		2.08	0.56	26.39	85.0	26.67	100.0	29.44	[illegible]	10.33	1.27
230		2.08	0.62	26.67	85.0	[illegible]	100.0	32.00	76.0	11.07	[illegible]
200	2	2.10	0.57	26.67	84.5	26.94	100.0	32.00	77.0	10.73	2.03
160		2.10	0.55	26.39	85.0	28.20	100.0	31.67	85.0	10.33	2.03
230		2.10	0.62	25.67	85.0	28.00	100.0	32.00	[illegible]	11.33	[illegible]
200	3	2.20	0.57	26.67	84.5	26.94	100.0	32.00	80.0	11.13	3.56
160		2.20	0.55	26.39	85.0	26.94	100.0	31.67	85.0	10.67	3.43
ID FAN											
230		2.12	0.45	28.33	82.0	28.33	100.0	30.83	91.5	6.80	0.76
200	1	2.20	0.41	27.78	86.0	28.89	100.0	31.11	90.5	6.33	0.51
160		2.25	0.39	27.78	84.5	29.17	100.0	31.39	91.0	6.33	0.51
230		2.10	0.45	26.67	90.5	27.22	100.0	30.83	90.0	6.20	1.27
200	2	2.15	0.42	26.67	93.0	27.78	100.0	31.39	84.0	6.13	1.11
160		2.20	0.39	26.67	92.0	27.78	100.0	32.22	[illegible]	5.73	1.02
230		2.05	0.44	27.80	87.0	28.33	99.0	31.94	81.5	6.00	2.27
200	3	2.15	0.40	[illegible]	85.0	[illegible]	96.5	33.60	80.5	5.67	2.16
160		2.15	0.38	28.06	84.0	28.33	92.0	31.94	85.0	5.33	2.03

TABLE 4.6

Psychrometric and Pressure drop data for CADE

Angle of Orientation : 90°, Air inlet area: 0.1006 m^2

Make up water: 0.518 kg/min., Water flow rate: 80 kg/min.

V (Volts)	No. of stages n	Compressor power (kW)	Fan power (kW)	Inlet conditions dbt t_1 (°C)	Inlet conditions ϕ_1 ( )	Discharge air without heater dbt t_2 (°C)	Discharge air without heater ϕ_2 (%)	Discharge air with heater dbt t_3 (°C)	Discharge air with heater ϕ_3 (%)	Velocity v (m/s)	Pressure drop Δp (mm)
FD FAN											
230		2.30	0.55	29.60	82.0	28.00	100.0	30.83	91.0	10.67	2.54
200	1	2.40	0.53	29.20	87.0	21.50	100.0	30.83	86.0	10.53	2.41
160		2.30	0.50	27.60	88.0	28.60	100.0	31.11	86.0	10.00	2.27
230		2.00	0.50	29.00	90.0	28.20	100.0	31.00	[illegible]	10.80	3.91
200	2	2.15	0.47	29.00	90.0	29.20	100.0	31.60	[illegible]	10.40	3.56
160		2.20	0.44	28.60	80.5	28.00	100.0	32.00	91.5	9.93	3.05
230		2.20	0.50	28.00	96.0	27.50	100.0	31.11	89.0	10.67	4.32
200	3	2.25	0.49	25.56	96.0	28.00	100.0	31.00	91.0	10.07	4.19
160		2.30	0.47	25.56	94.0	27.22	100.0	31.11	95.0	10.07	4.06
ID FAN											
230		2.30	0.47	26.67	89.0	28.06	100.0	31.67	83.5	6.20	1.40
200	1	2.40	0.43	26.67	[illegible]	28.33	100.0	31.94	84.5	5.87	1.35
160		2.35	0.40	26.67	92.5	28.00	100.0	32.00	81.0	5.73	1.35
230		2.25	0.45	26.67	90.0	27.22	100.0	32.22	77.0	5.80	2.29
200	2	2.30	0.45	26.94	90.0	27.78	100.0	32.50	83.5	5.87	2.16
160		2.22	0.36	26.94	90.0	27.78	100.0	32.78	83.0	5.73	2.03
230		2.27	0.44	26.94	88.0	28.61	[illegible]	33.06	78.0	5.40	3.05
200	3	2.30	0.40	27.50	86.0	28.33	98.0	31.94	80.5	5.27	2.92
160		2.22	0.37	28.33	81.0	28.61	96.5	32.78	[illegible]	5.07	2.27

TABLE 4.7

Psychrometric and Pressure drop data for CDE with FD Fan, Air inlet area: 0.1006 m^2
Make up water: 0.518 kg/min., Water flow rate: 80 kg/min.

V (Volts)	Angle θ (degrees)	Compressor power (kW)	Fan power (kW)	Inlet conditions dbt t_1 (°C)	Inlet conditions ϕ_1 (%)	Discharge air without heater dbt t_2 (°C)	Discharge air without heater ϕ_2 (%)	Discharge air with heater dbt t_3 (°C)	Discharge air with heater ϕ_3 (%)	Velocity v (m/s)	Pressure drop Δp (mm)
230		2.10	1.02	26.00	81.0	26.11	100.0	30.00	92.5	11.33	1.79
200	15	2.10	0.74	26.00	81.0	26.67	100.0	29.72	95.0	11.00	1.65
160		1.90	0.45	26.00	83.0	26.00	100.0	30.00	85.5	10.53	1.50
230		2.15	0.62	26.11	80.0	26.11	100.0	30.00	92.0	11.20	2.16
200	30	2.22	0.57	26.11	80.0	26.11	100.0	30.00	87.0	10.80	20.3
160		2.20	0.55	26.11	81.0	26.11	100.0	30.56	83.0	10.47	1.78
230		2.22	0.64	25.58	80.5	26.00	100.0	30.83	81.5	11.20	2.67
200	45	2.30	0.55	25.56	85.0	25.56	100.0	30.00	81.0	10.80	2.41
160		2.25	0.55	25.56	85.0	25.90	100.0	30.83	77.0	10.33	2.03
230		2.00	0.61	26.39	79.0	26.39	100.0	31.11	82.0	11.13	4.57
200	60	2.05	0.55	26.50	92.0	26.67	100.0	30.83	82.0	10.47	4.45
160		2.00	0.54	26.50	82.0	26.67	100.0	31.67	82.0	10.27	4.06
230		2.22	0.62	26.00	81.0	25.83	100.0	31.67	80.0	11.33	4.57
200	75	2.42	0.55	26.00	84.0	26.67	100.0	32.00	79.0	11.00	3.80
160		2.30	0.55	26.00	84.0	26.67	100.0	30.00	36.0	10.27	3.56
230		1.87	1.03	23.61	74.0	21.67	100.0	26.35	75.0	11.93	3.30
200	90	1.90	0.73	24.44	71.5	23.33	99.5	26.67	83.0	11.00	2.67
160		1.87	0.46	23.89	73.5	22.78	100.0	26.67	80.0	10.80	2.54

TABLE 4.8

Psychrometric and pressure drop data for CDE with ID Fan , Air inlet area: 0.1006 m^2
Make up water: 0.518 kg/min., Water flow rate: 80 kg/min.

V (Volts)	Angle θ (degrees)	Compressor power (kW)	Fan power (kW)	Inlet conditions dbt t_1 (°C)	Inlet conditions ϕ_1 (%)	Discharge air without heater dbt t_2 (°C)	Discharge air without heater ϕ_2 (%)	Discharge air with heater dbt t_3 (°C)	Discharge air with heater ϕ_3 (%)	Velocity v (m/s)	Pressure drop Δp (mm)
230		2.10	0.83	25.83	77.5	25.83	94.0	30.56	80.0	0.38	6.00
200	15	2.10	0.53	25.00	79.0	25.00	95.5	30.83	75.0	0.13	5.73
160		1.55	0.32	25.00	79.0	25.00	95.5	30.00	79.0	0.13	5.60
230		2.05	0.46	25.60	79.0	25.83	95.0	31.94	76.0	0.76	5.47
200	30	2.20	0.42	25.50	79.0	26.67	93.5	32.50	73.5	0.66	5.33
160		2.20	0.40	25.50	79.0	26.00	94.0	32.22	74.0	0.51	5.33
230		2.05	0.47	26.00	82.5	26.39	96.5	32.50	74.0	1.27	6.00
200	45	2.20	0.42	25.56	82.5	26.67	95.5	32.78	71.0	1.14	5.73
160		2.10	0.41	25.56	82.5	26.00	96.5	32.50	69.0	1.02	5.47
230		1.97	0.43	26.11	82.5	26.39	97.5	32.00	74.0	2.03	5.73
200	60	2.00	0.40	25.56	81.0	26.67	96.0	32.22	77.0	2.03	5.33
160		2.00	0.37	25.00	85.0	26.39	97.5	32.50	70.0	1.91	5.20
230		2.05	0.42	26.39	80.5	26.39	93.5	32.00	70.0	1.78	5.67
200	75	2.05	0.40	25.56	79.0	26.11	93.0	32.22	68.0	1.53	5.47
160		1.95	0.37	26.11	80.0	26.67	94.0	32.50	69.5	1.40	5.33
230		1.95	0.94	23.61	77.0	24.44	93.0	31.11	74.0	1.53	5.47
200	90	2.00	0.55	23.06	76.5	24.11	93.0	31.11	72.0	1.40	5.33
160		2.00	0.35	23.33	74.0	24.44	94.0	30.56	73.5	1.40	5.13

TABLE 4.9

Psychrometric and Pressure drop data for HDE with FD Fan , Air inlet area: 0.1006 m^2

Make up water: 0.518 kg/min., Water flow rate: 80 kg/min.

V (Volts)	No. of stages n	Compressor power (kW)	Fan power (kW)	Inlet conditions dbt t_1 (°C)	ϕ_1 (%)	Discharge air without heater dbt t_2 (°C)	ϕ_2 (%)	Discharge air with heater dbt t_3 (°C)	ϕ_3 (%)	Velocity v (m/s)	Pressure drop Δp (mm)
230		1.85	1.07	23.89	78.0	22.78	100.0	26.67	88.0	12.13	0.83
200	3	1.83	0.74	24.44	75.0	22.83	100.0	26.94	86.0	11.44	0.38
160		1.80	0.49	23.89	78.0	23.06	100.0	26.74	86.0	11.00	0.25
140		1.80	0.46	24.44	75.0	23.06	100.0	26.67	87.0	10.33	0.12
230		1.80	1.08	21.94	74.0	21.11	100.0	25.00	90.0	12.00	0.50
200	2	1.85	0.71	22.00	76.0	21.11	100.0	25.00	89.5	11.67	0.38
160		1.90	0.47	22.00	80.5	21.11	100.0	25.00	90.0	11.47	0.33
140		1.92	0.45	22.00	76.0	21.50	100.0	25.00	90.0	10.82	0.12
230		1.81	1.05	21.35	74.0	21.11	100.0	26.67	94.0	12.22	0.12
200	1	1.78	0.69	21.94	74.0	21.11	100.0	25.00	90.0	11.50	0.07
160		1.80	0.44	21.39	74.0	21.11	95.0	26.67	79.5	11.24	0.00
140		1.74	0.42	22.00	76.0	21.10	100.0	25.00	89.5	10.94	0.00

TABLE 4.10

Psychrometric and Pressure drop data for HDE with ID Fan
Air inlet area: 0.1006 m^2, Make up water: 0.518 kg/min.
Water flow rate: 80 kg/min.

V (Volts)	No. of stages n	Compressor Power (kW)	Fan power (kW)	Inlet conditions dbt t_1 (°C)	Inlet conditions ϕ_1 (%)	Discharge air without heater dbt t_2 (°C)	Discharge air without heater ϕ_2 (%)	Discharge air with heater dbt t_3 (°C)	Discharge air with heater ϕ_3 (%)	Velocity v (m/s)	Pressure drop Δp (mm)
230		1.87	0.87	24.44	64.0	22.22	85	30.56	66.2	6.07	0.00
200	1	1.88	0.34	24.00	69.0	23.89	82	28.61	70.5	5.83	0.00
160		1.89	0.35	24.00	64.0	23.89	78	30.28	67.0	5.60	0.00
140		1.92	0.59	23.33	66.0	23.89	79	30.56	67.0	5.67	0.00
230		1.73	0.90	23.06	74.0	23.33	93	28.61	70.0	5.67	0.13
200	2	1.80	0.61	23.06	73.5	23.33	93	28.60	73.0	5.58	0.12
160		1.77	0.36	23.06	73.5	23.33	93	28.06	80.5	5.53	0.07
140		1.75	0.33	23.06	71.5	23.33	91	28.06	70.5	5.33	0.00
230		1.89	0.86	21.39	74.0	21.10	96	26.67	70.5	5.73	0.25
200	3	1.90	0.56	21.94	71.0	21.11	86	25.54	78.0	5.63	0.25
160		1.92	0.34	21.67	75.0	21.11	96	25.83	82.0	5.52	0.12
140		1.89	0.34	21.94	71.0	21.11	93	25.56	80.0	5.40	0.07

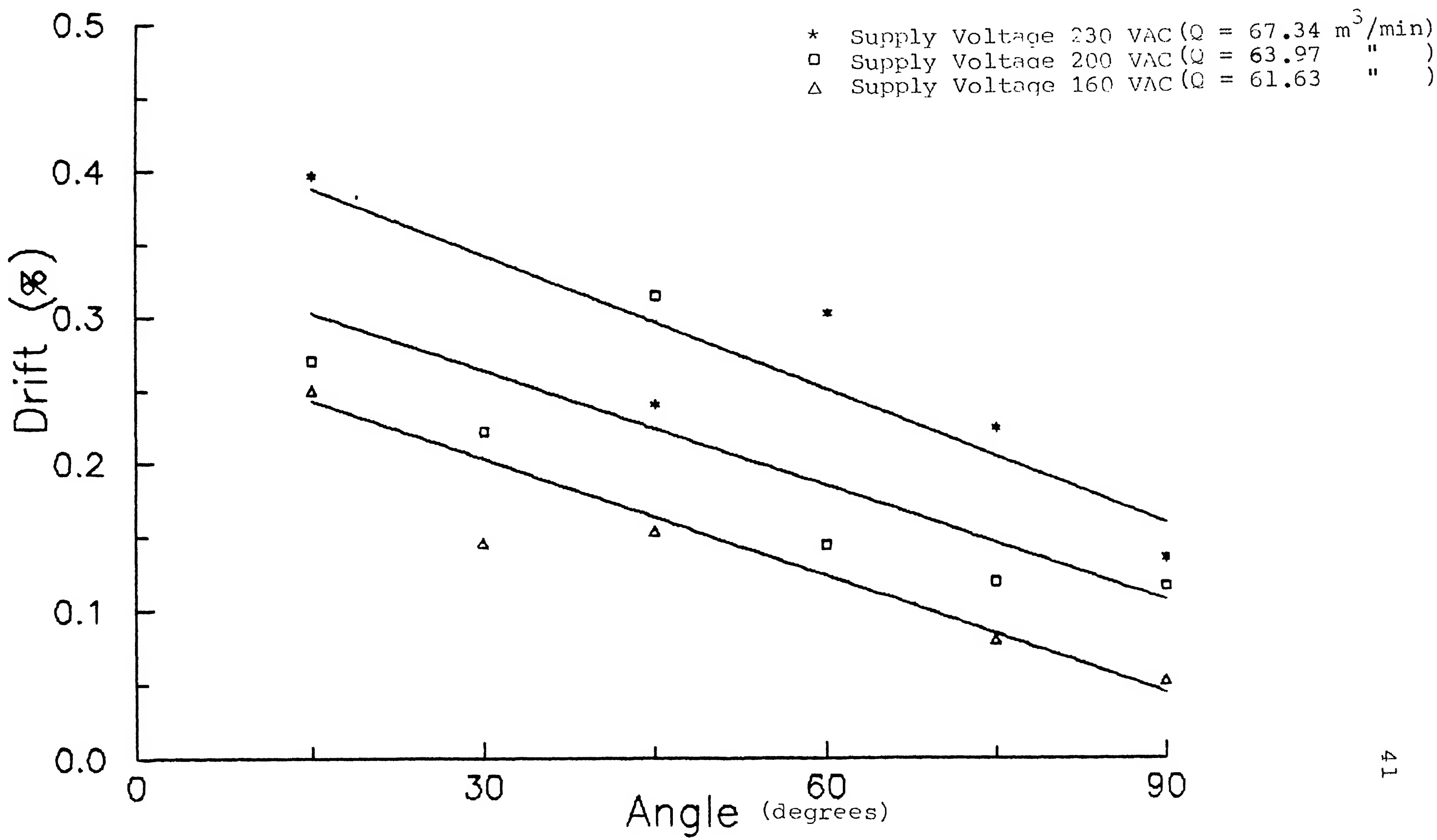


Fig: 4.1
DRIFT(%) vs Inclination angle for FD fan with single stage CA DE

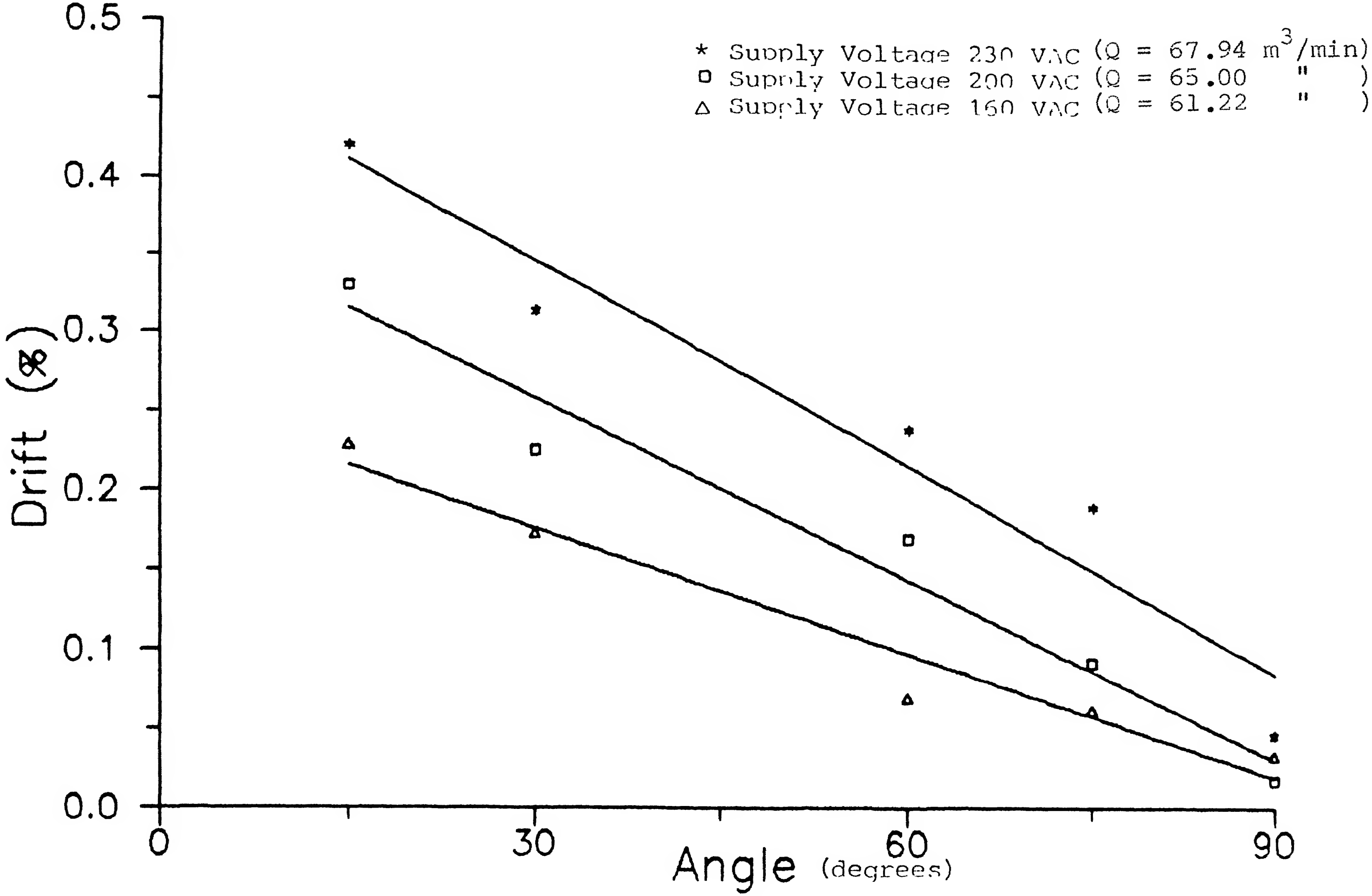
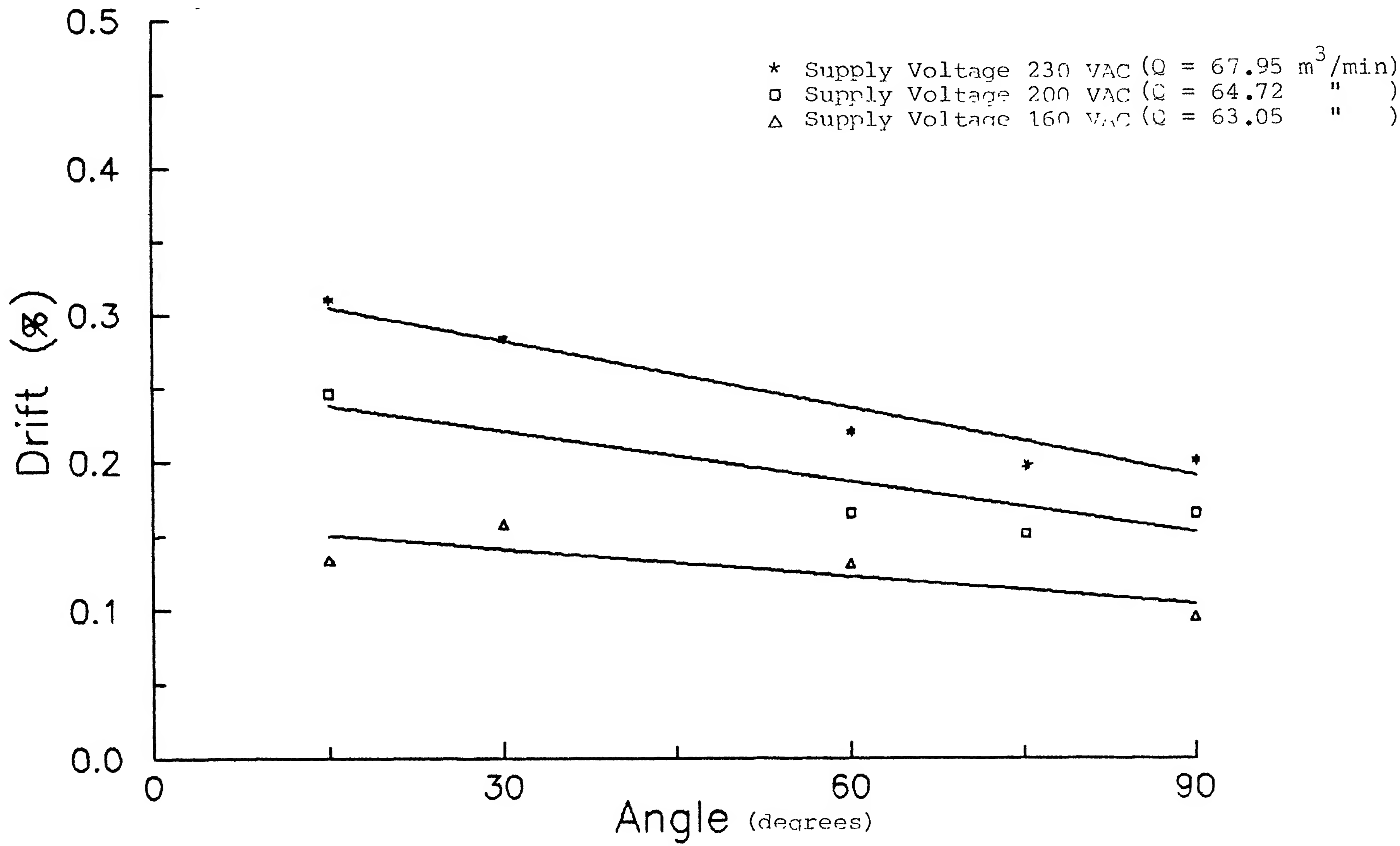


Fig: 4.2
DRIFT(%) Vs Inclination angle for FD fan with two stage CA DE



Drift Vs Inclination angle for FD fan with 3 stage CA DE
Fig: 4.3

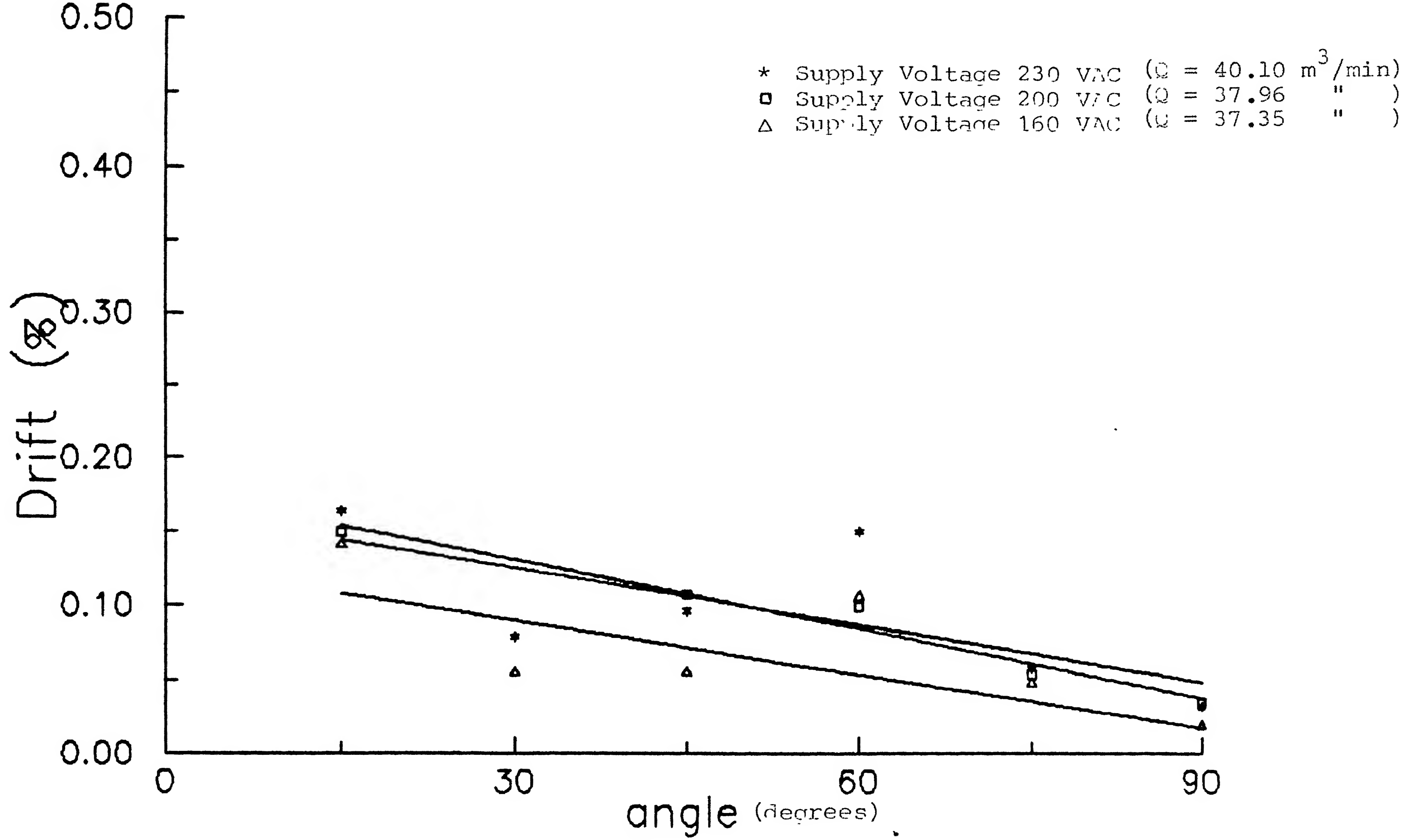
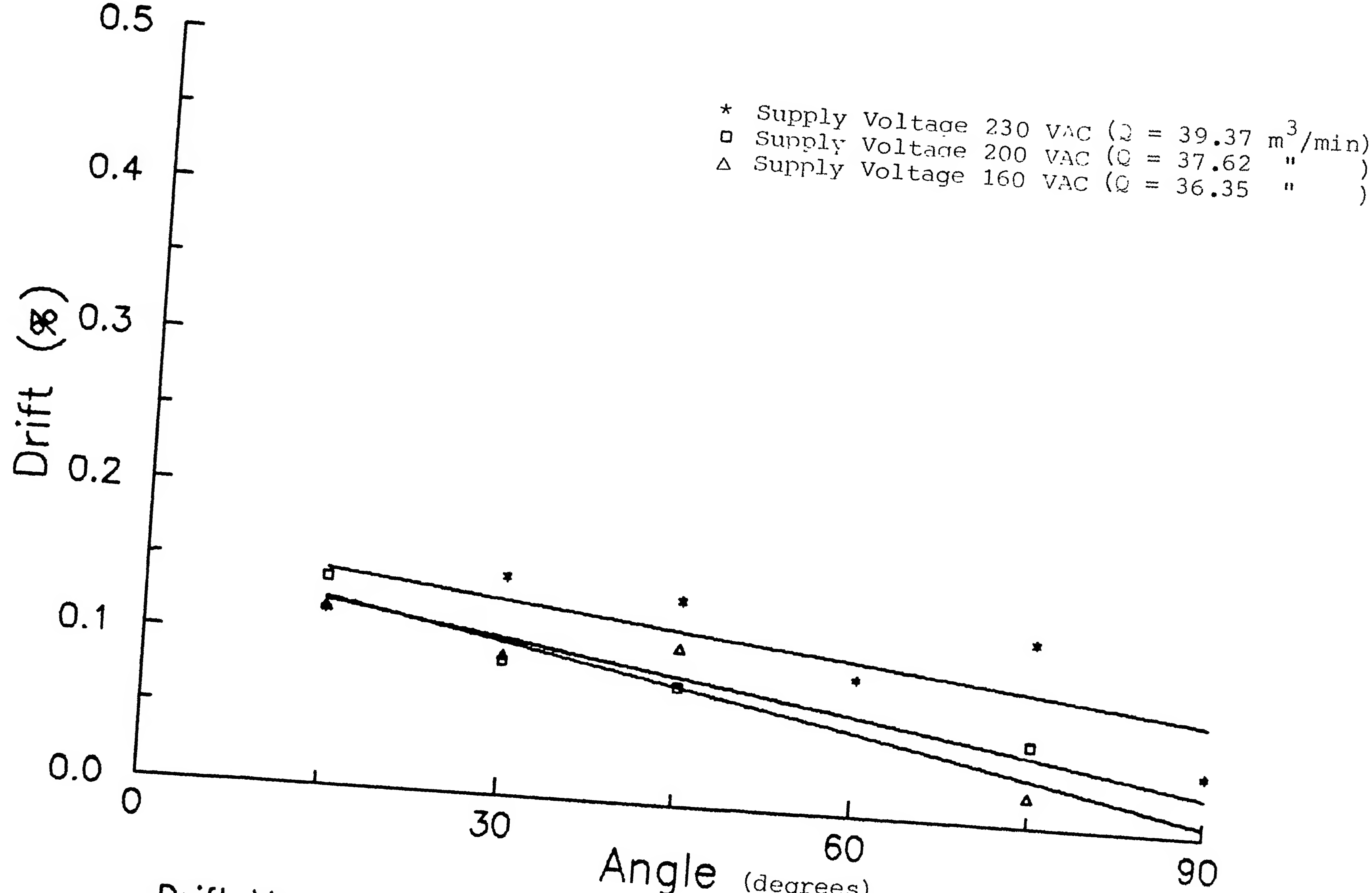


Fig: 4.4.
DRIFT(%) vs Inclination angle for ID fan with single stage CA DE



Drift Vs Inclination angle for ID fan with 2 stage CA DE
Fig: 4.5

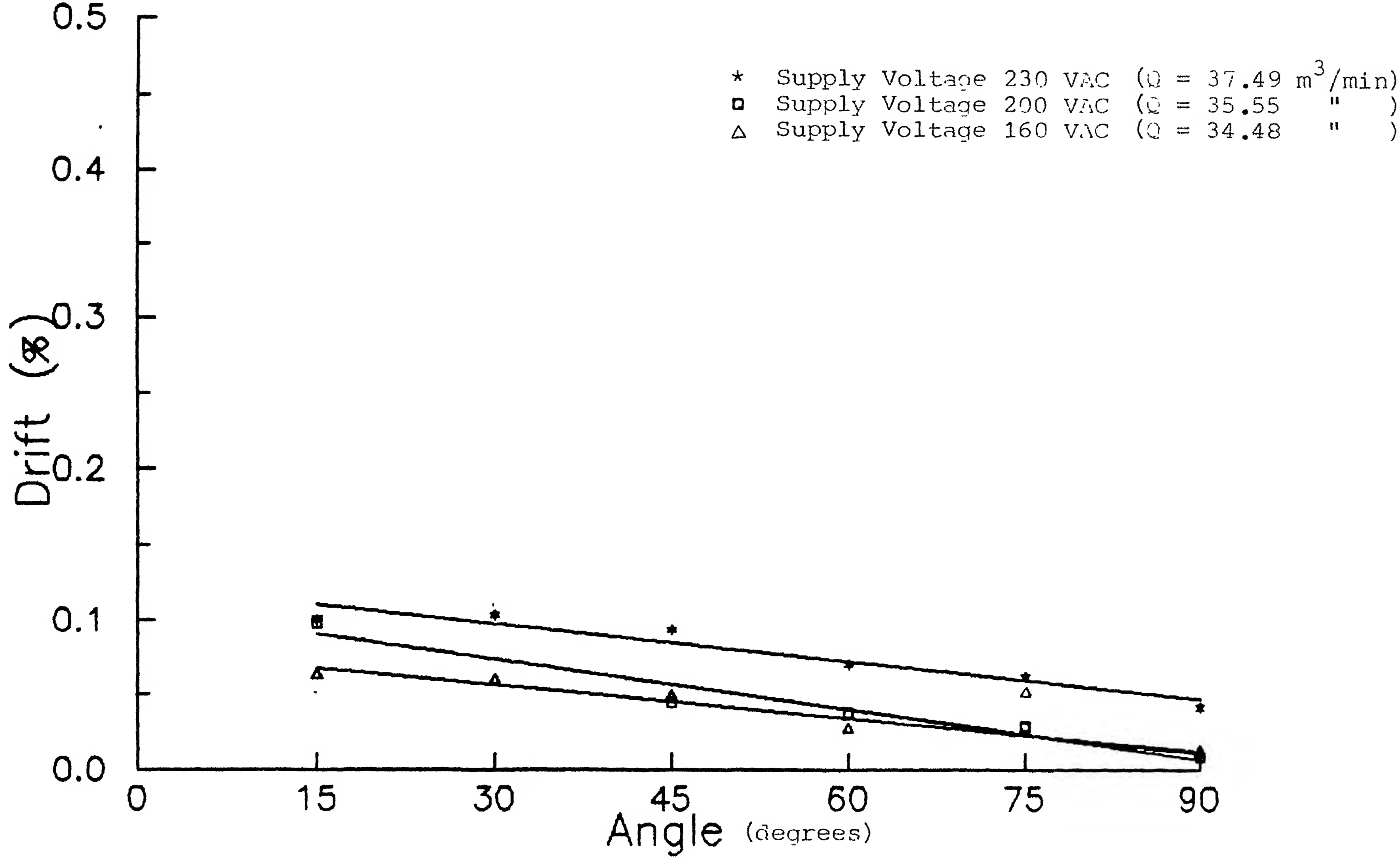
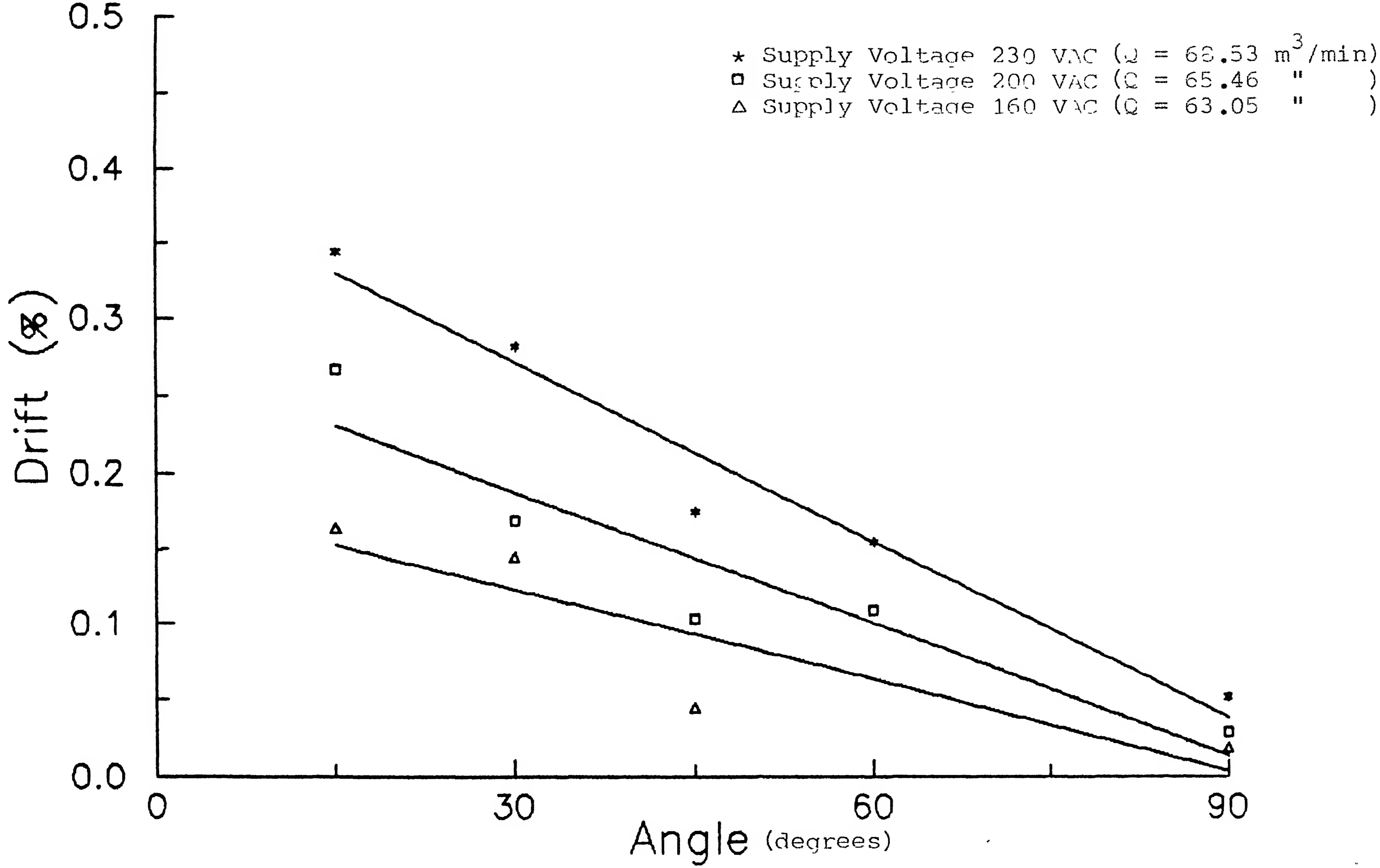
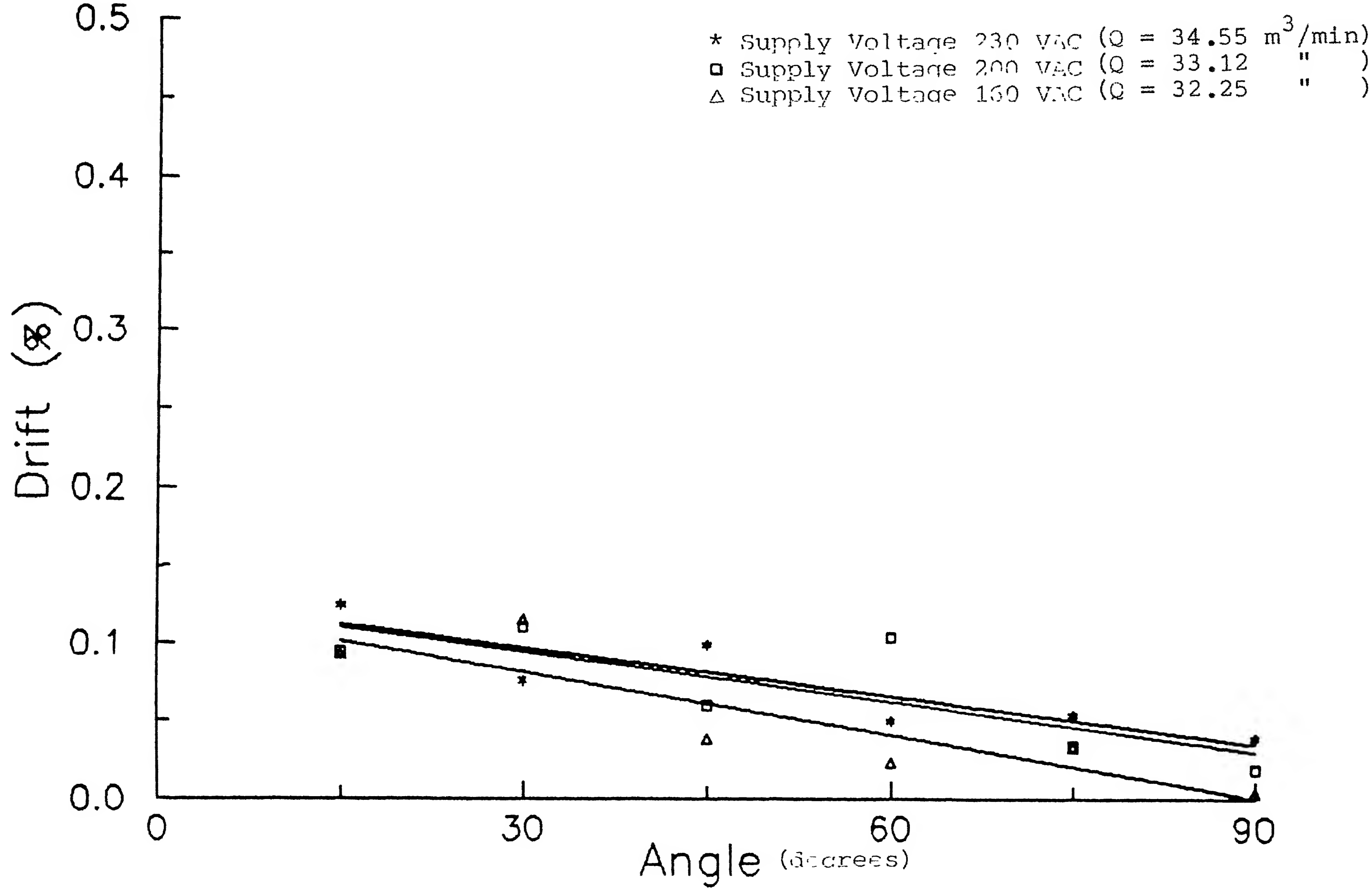


Fig: 4.6
DRIFT(%) vs Inclination angle for ID fan with three stage CA DE



Drift Vs Inclination angle for FD fan with 3 stage CONC DE
Fig: 4.7



Drift Vs Inclination angle for ID fan with 3stage CONC DE
Fig: 4.8

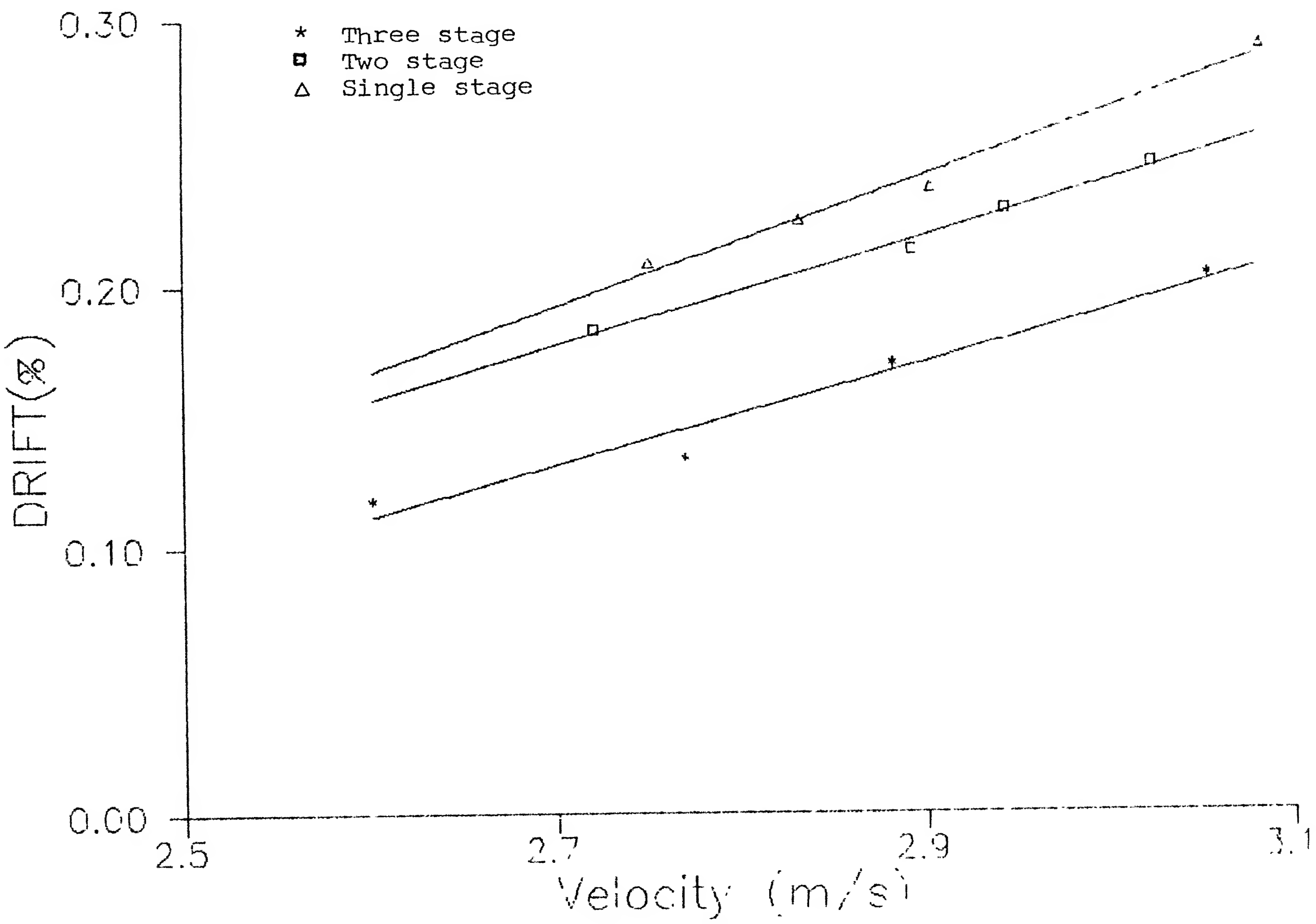


Fig: 4.9
DRIFT(%) vs Velocity for FD Fan with HDE

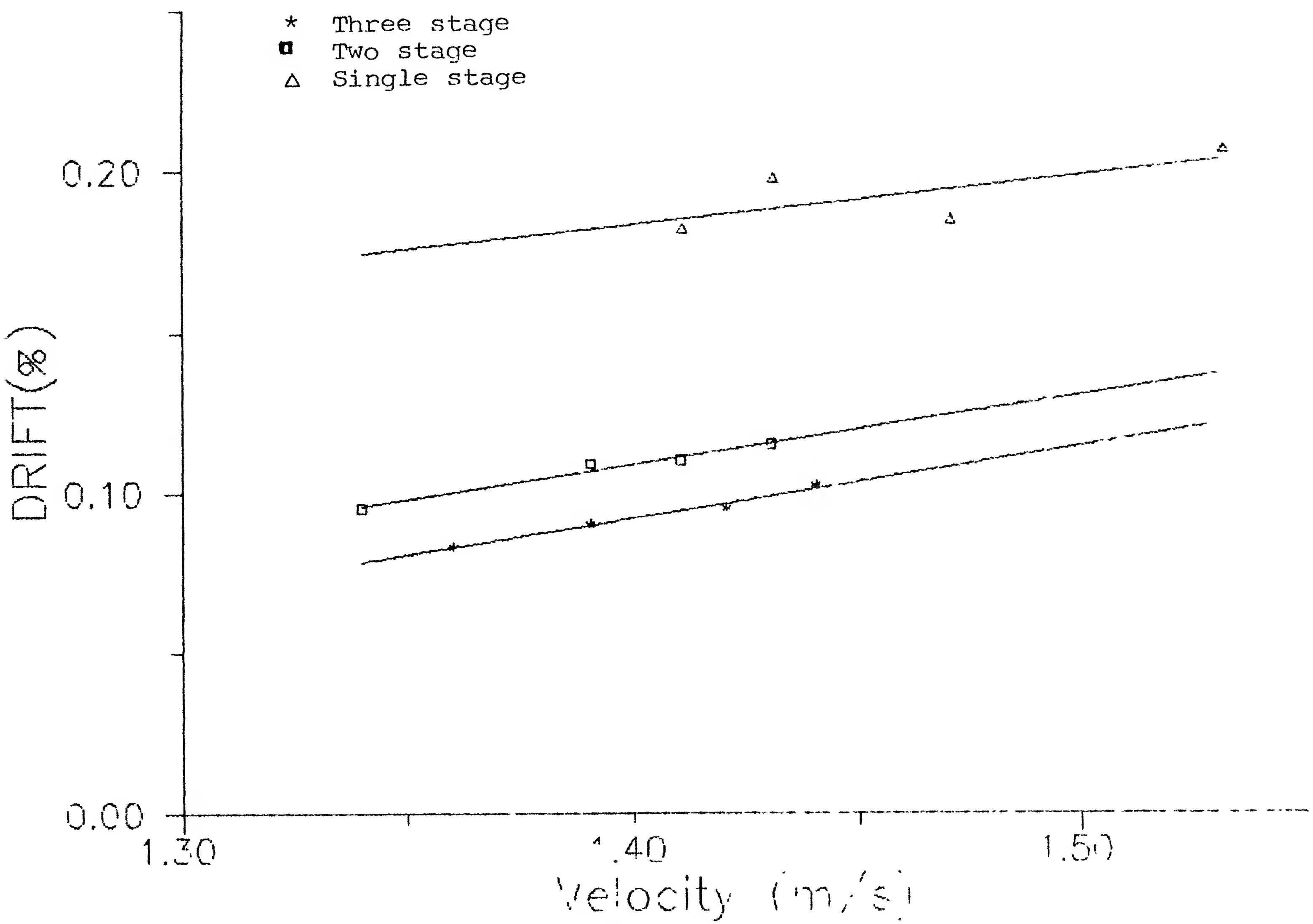


Fig: 4.10
DRIFT(%) vs Velocity for ID Fan with HDE

at a time were one, two or three. Each set of experiments was performed both for FD and ID fans separately. These experiments were repeated for both types of drift eliminators.

The variation of drift loss (expressed as a percentage of the circulating water flow rate) versus θ are shown in Figures 4.1 through 4.8 . As can be seen, the trend of these curves is similar. With an increasing angle of orientation, the drift loss decreases. The drift loss also decreases with an increase in the number of stages(n) of the drift eliminators.This is basically due to the fact as n increases or θ increases, the the static pressure drop across the drift eliminator stages increases. This in turn leaves a smaller fraction of the static pressure available for causing the flow, which results in a relatively small volumetric discharge. Thus the amount of water droplets carried along with the exit stream decreases. Besides as n increases, the exit air stream carrying the drift droplets makes a larger number of turns and thus the droplets undergo a more effective inertial separation.

The effect of increasing n can be seen for CADE from the Figures 4.1, 4.2, 4.3 for the forced flow and Figures 4.4, 4.5, 4.6 for the induced flow. One can also see from any one of these figures the effect of changing the fan speed by changing the supply voltage. As the fan speed is reduced, the volumetric discharge rate decreases and as expected the amount of drift also decreases. Each fan speed has a maximum value of drift loss around $\theta = 15^{o}$ and this maximum increases as the fan is increased. This may be attributed to a small pressure drop around $\theta = 15^{o}$ because of the avaibility of substantially large flow area. Corresponding to a

typical value of $\theta = 45^{o}$ it can be seen that the drift consistently reduces as the number of stages is increased.

For HDE, the drift loss was plotted versus the velocity of the air through the DE for one, two and three stages, both for forced and induced flows. The typical curves are shown in Figures 4.9 and 4.10. The drift loss decreases as n increases, due to a larger pressure drop across the DE stages and also because of relatively more effective inertial separation Also decreases with with a decrease in the flow velocity or the flow rate of air.

4.3 Pressure Drop

The pressure drop, Δp, was recorded for θ varying between 15^{o} to 90^{o} for CADE and CDE. For each value of θ, the supply voltage was varied in the range of 230 VAC to 160 Vac in order to change the fan speed or the air flow rate. The data are given in Tables 4.1 through 4.6 for CADE and Tables 4.7 and 4.8 for CDE. The pressure drop versus θ is plotted in Figures 4.11 through 4.18 for CADE and CDE.

From these curves it is inferred that as θ increases, Δp also increases due to a reduction in the available flow area. With an increase in fan speed, the fan discharge and therefore the flow velocity increases resulting in a larger pressure drop. As n increases, Δp increases due to a larger resistance to the flow. The pressure drop for practical values of $\theta(45^{o}- 60^{o})$ is smaller for the CADE

For HDE the pressure drop was plotted versus the velocity of the air through the DE in the Figures 4.19 and 4.10. The pressure

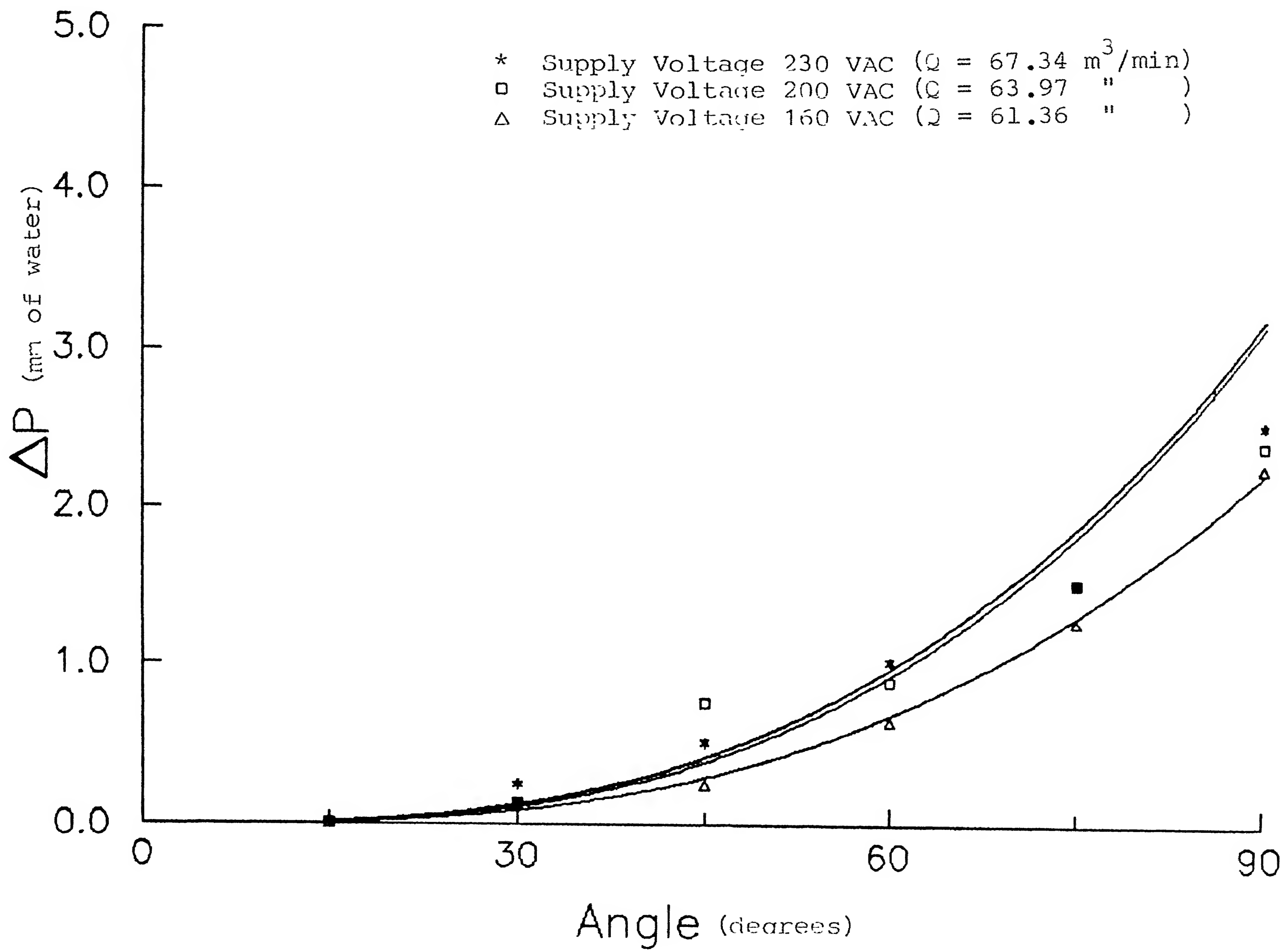


Fig: 4.11
ΔP vs Inclination Angle for FD Fan with Single stage CA DE

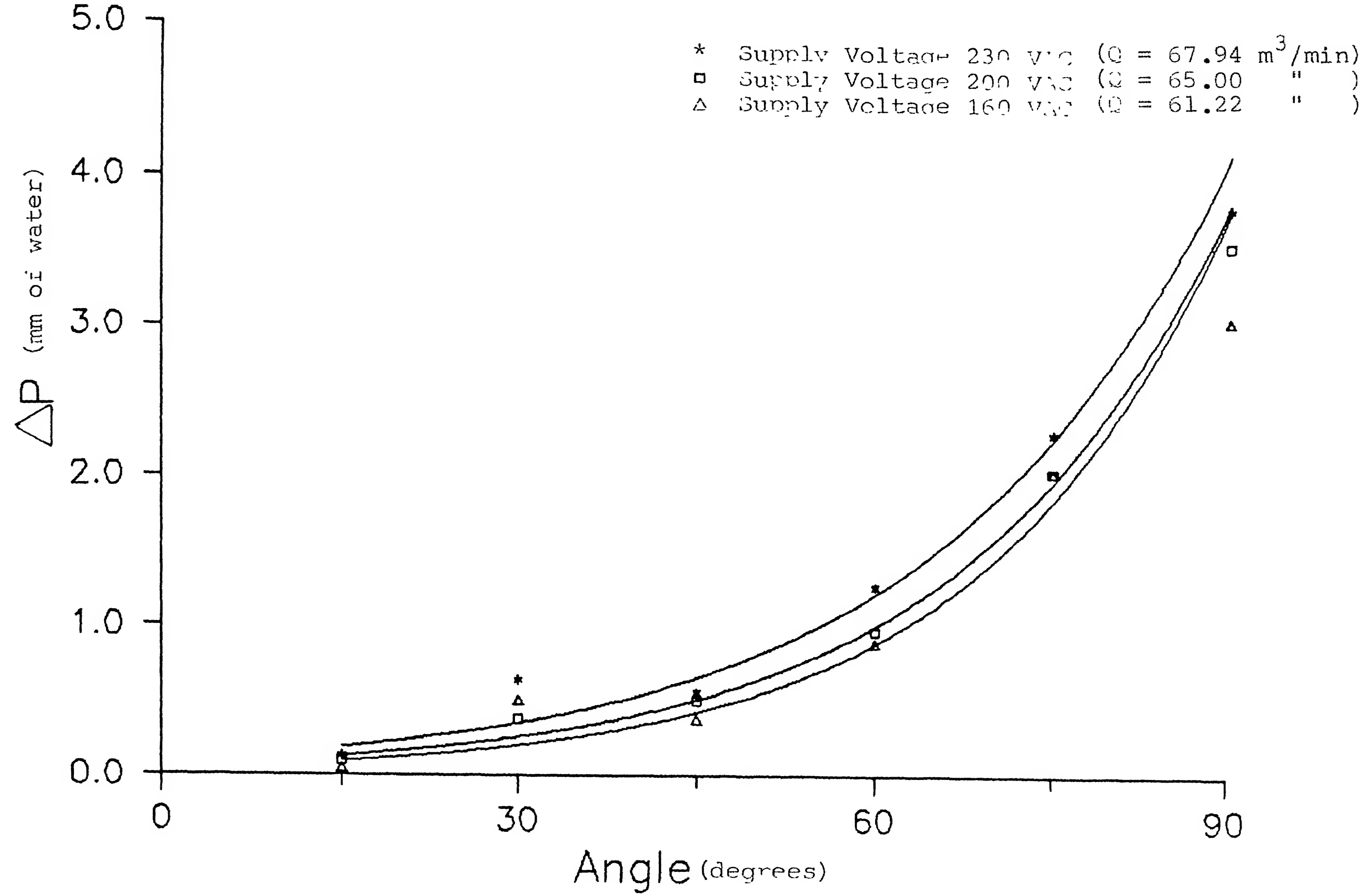


Fig: 4.12
ΔP vs Inclination Angle for FD Fan with 2 stage CA DE

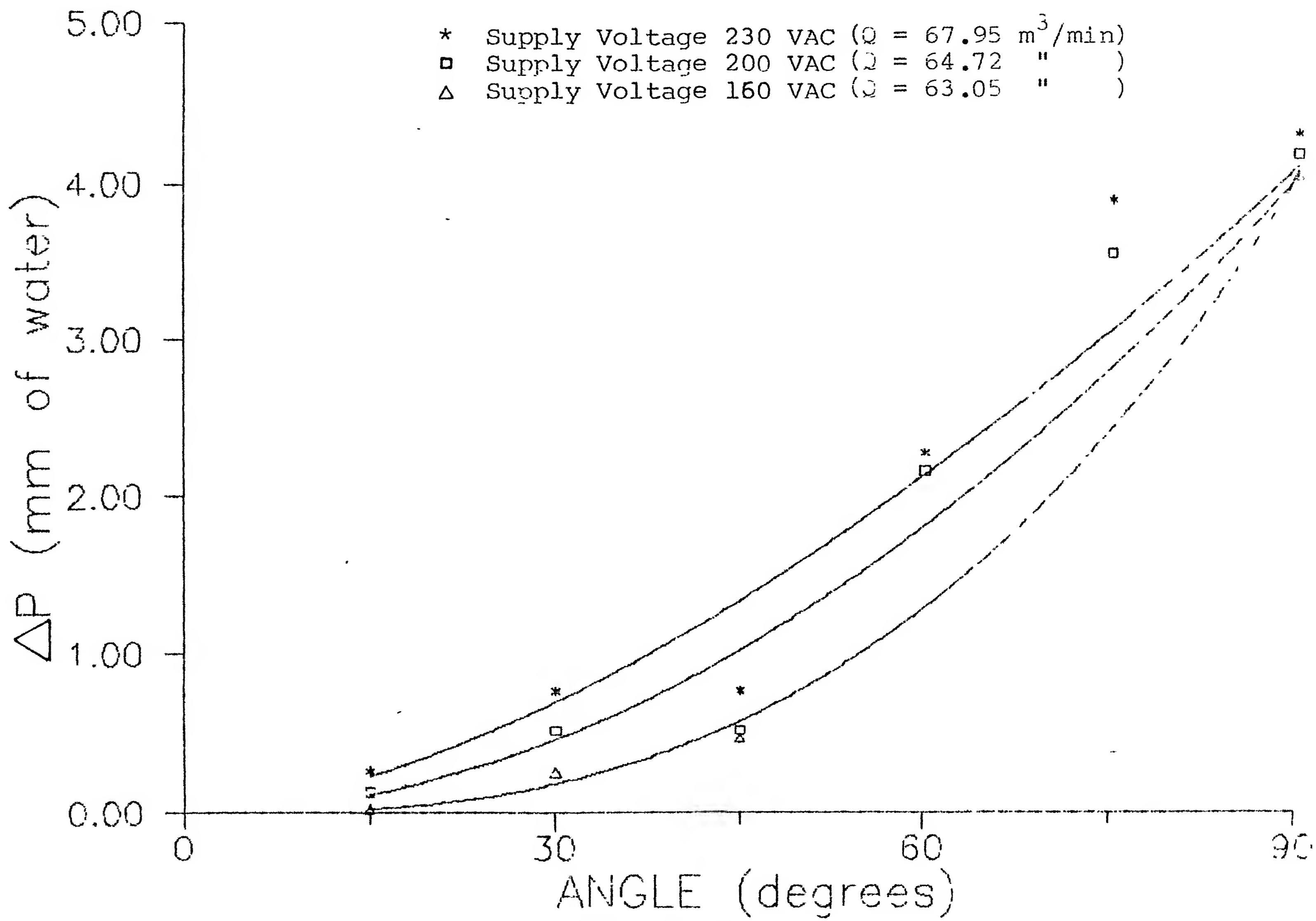


Fig: 4.13

ΔP vs Inclination Angle for FD fan with three stage CA DE

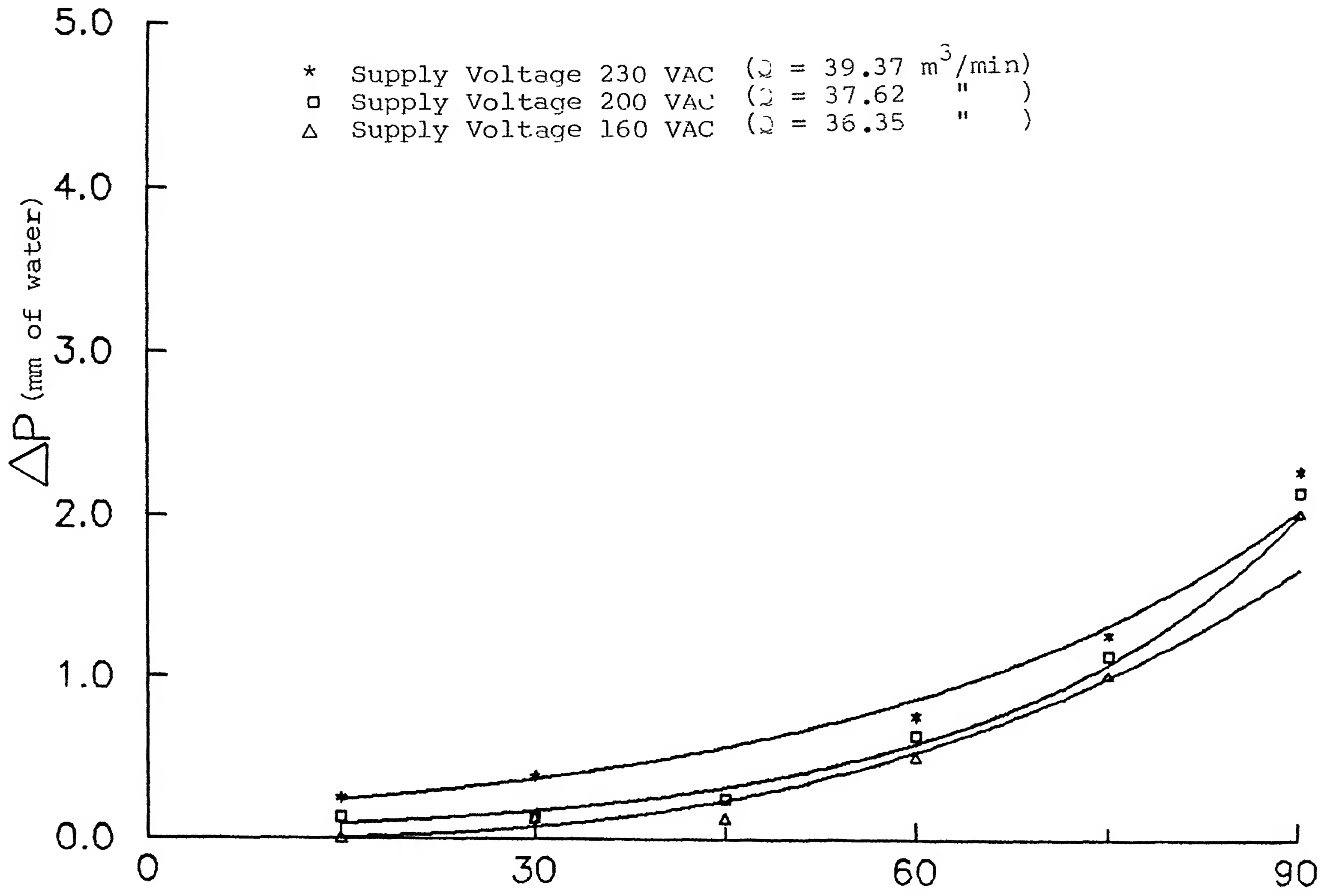


Fig: 4.15
ΔP vs Inclination Angle for ID Fan with 2 stage CA DE

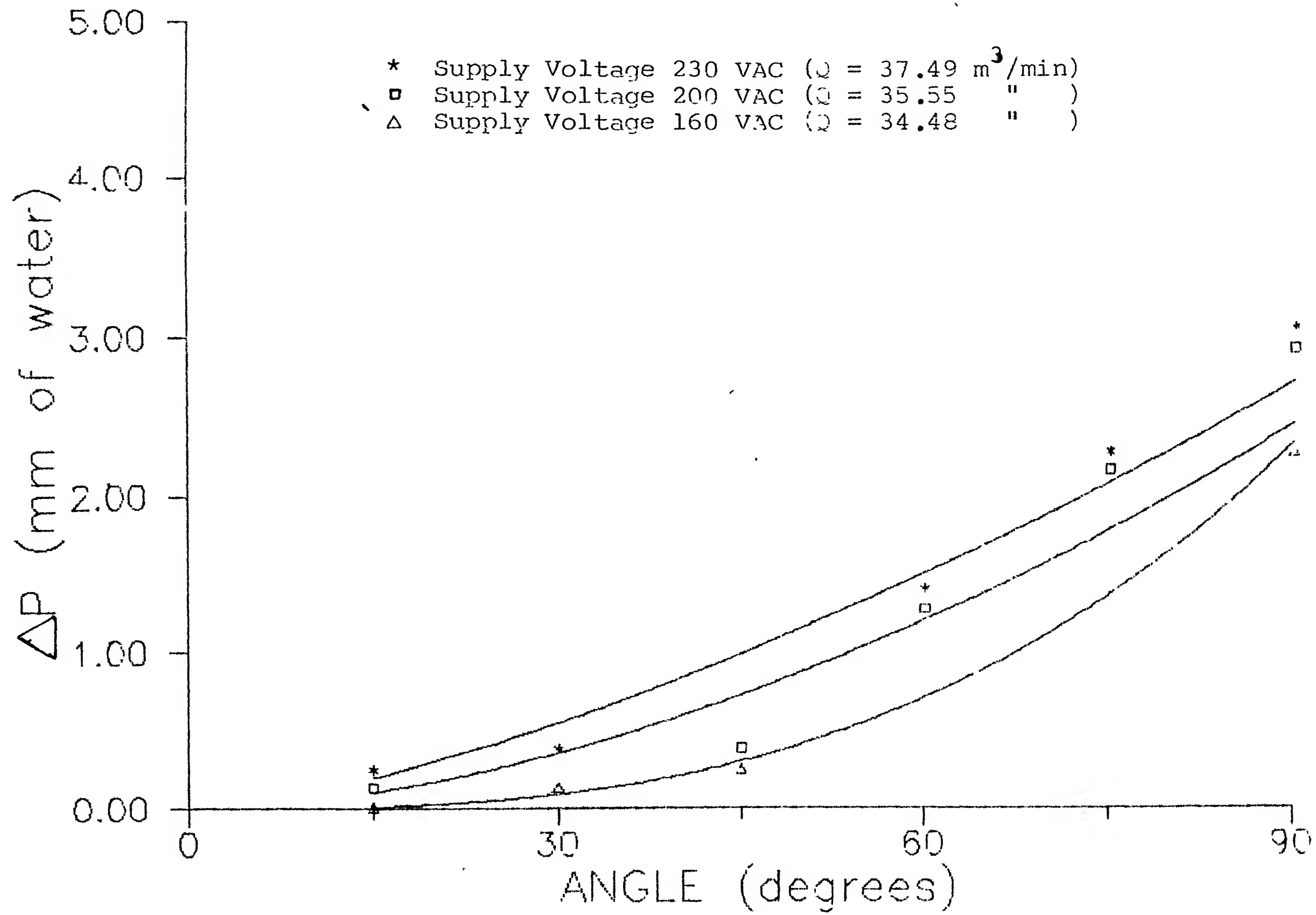


Fig: 4.16
ΔP vs Inclination Angle for ID fan with three stage CA DE

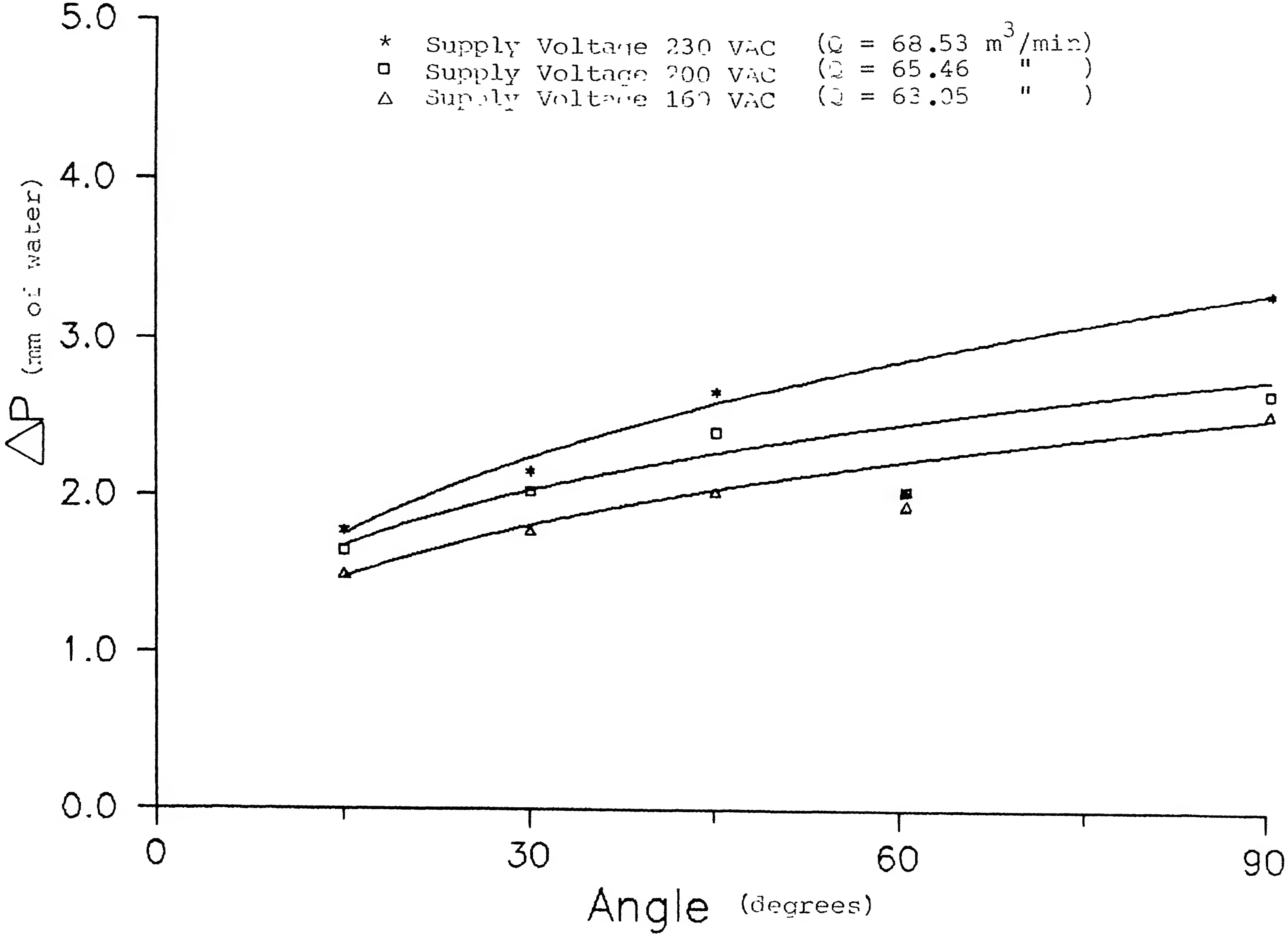


Fig: 4.17
ΔP vs Inclination Angle for FD Fan with 3 stage CONC DE

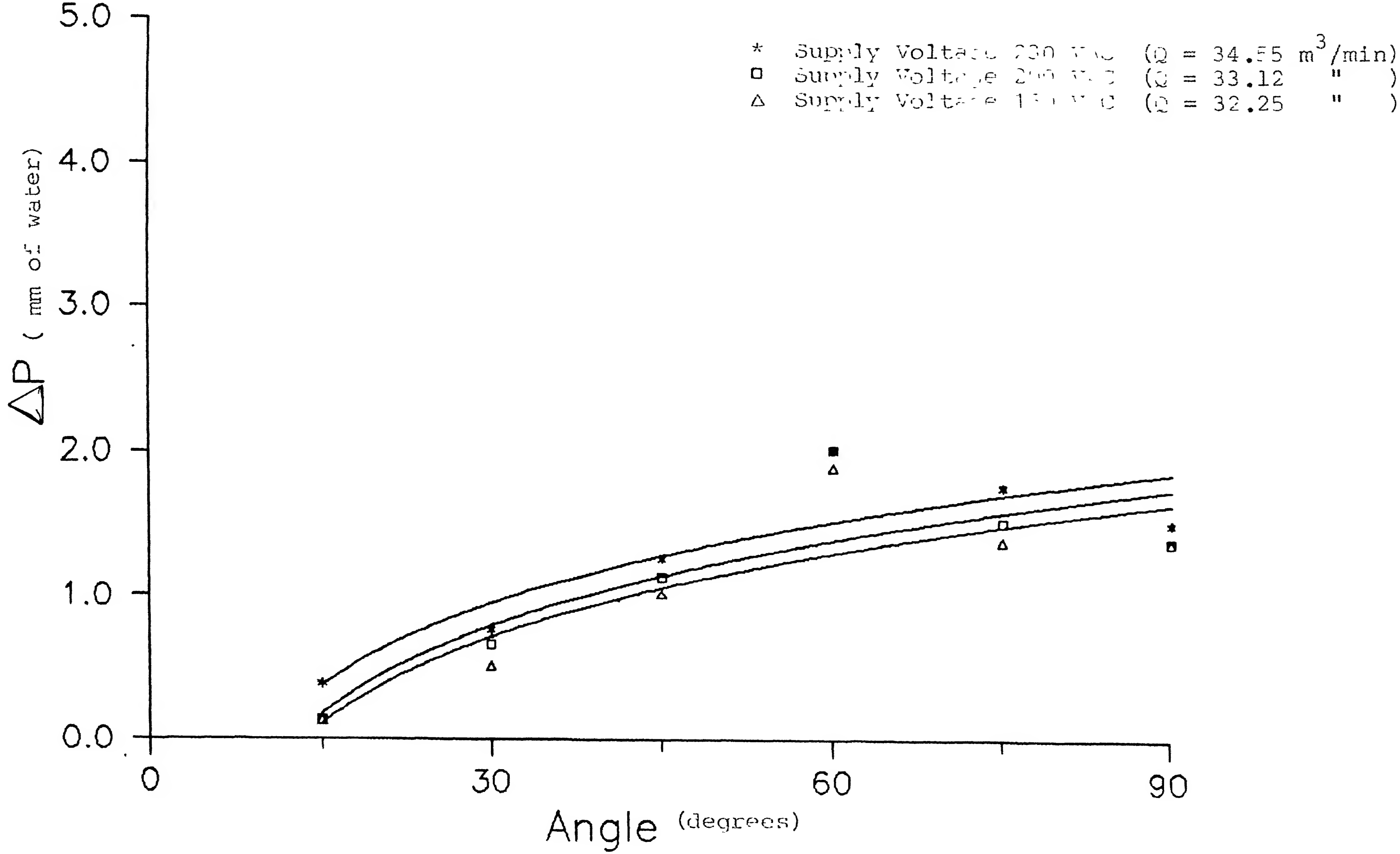


Fig: [illegible].18
ΔP vs Inclination Angle for ID Fan with 3 stage CONC DE

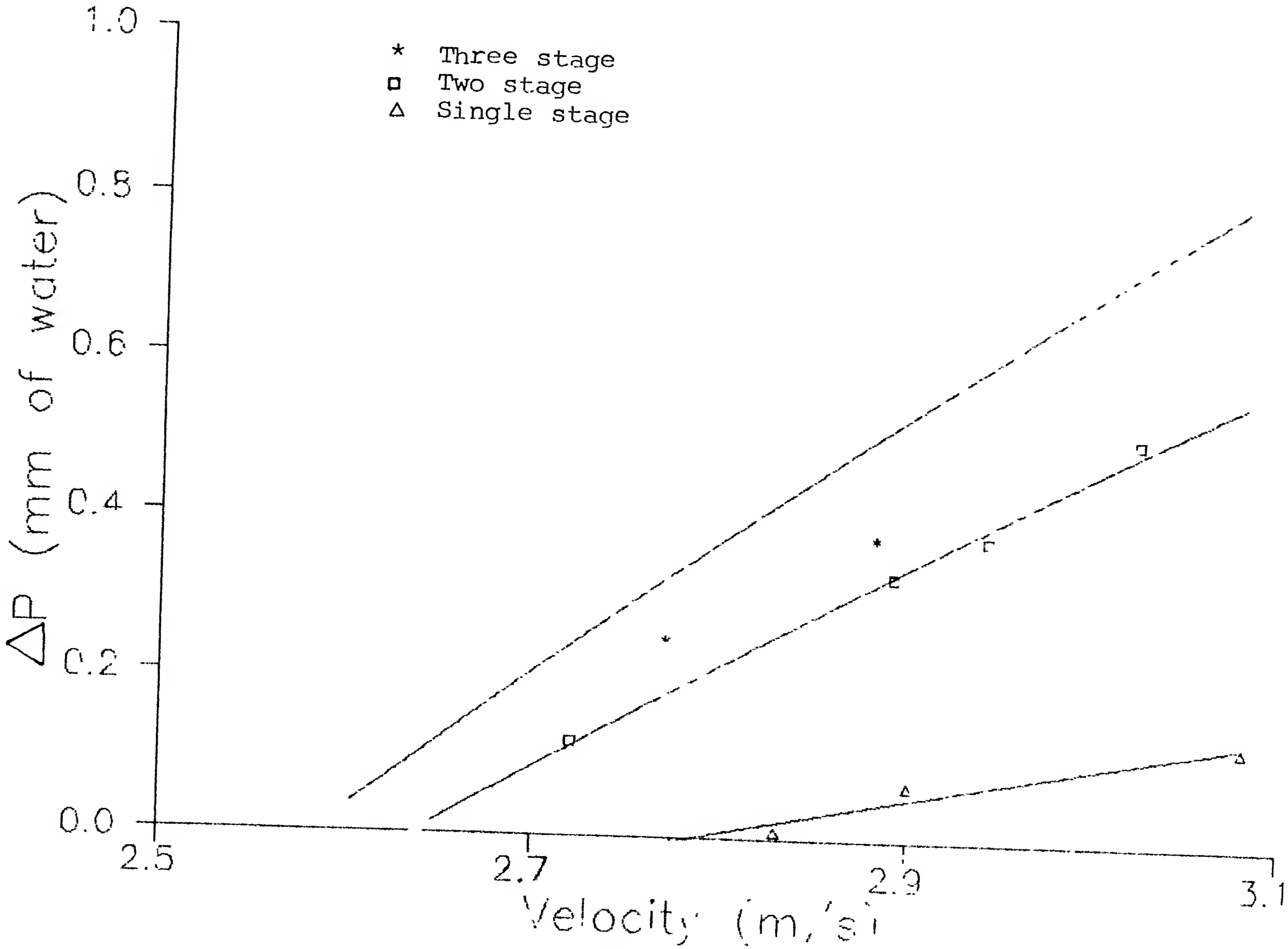


Fig: 4.19
ΔP vs Velocity for HDE with FD Fan

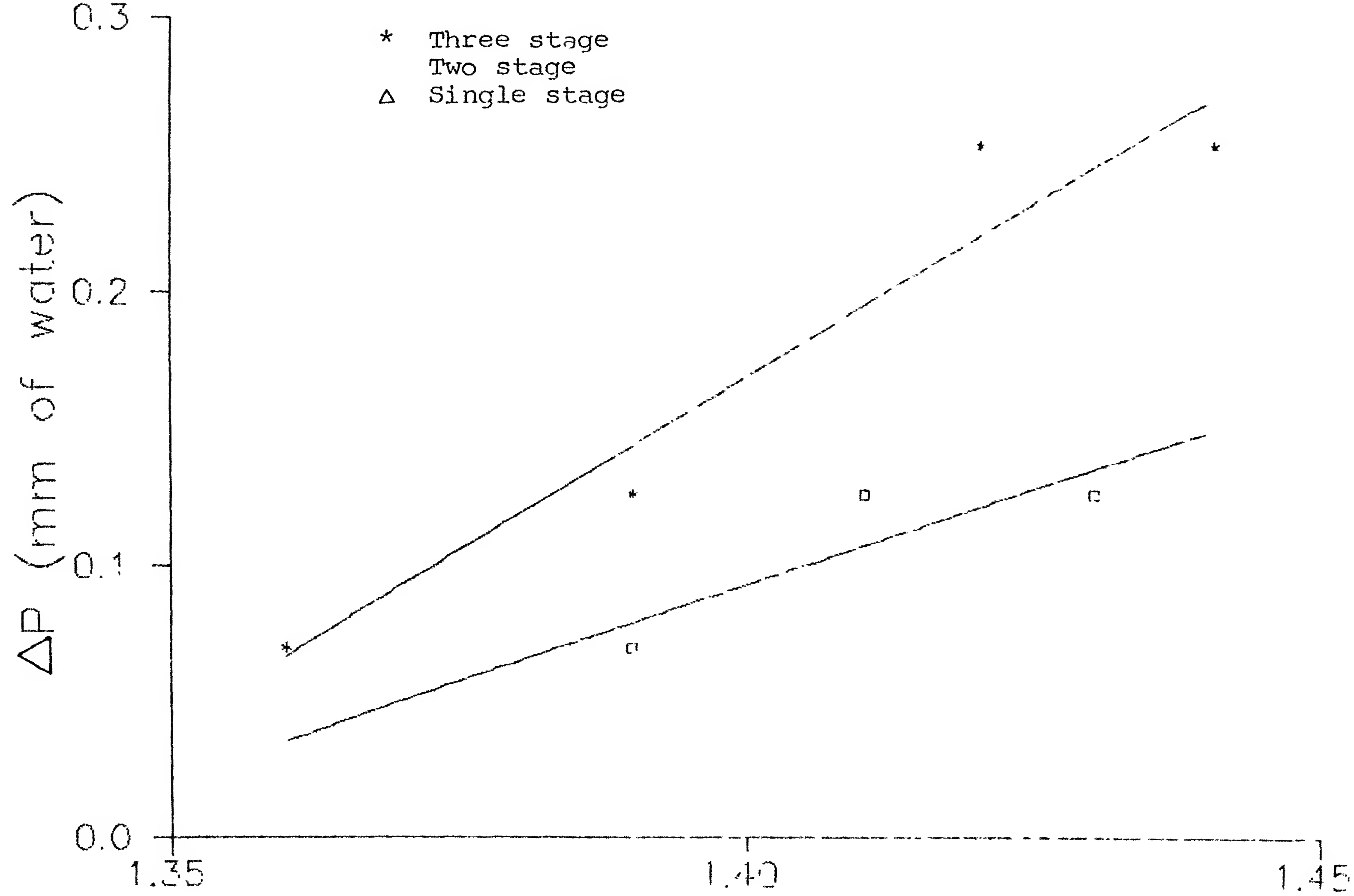


Fig: 4.20
ΔP vs Velocity for HDE with ID Fan

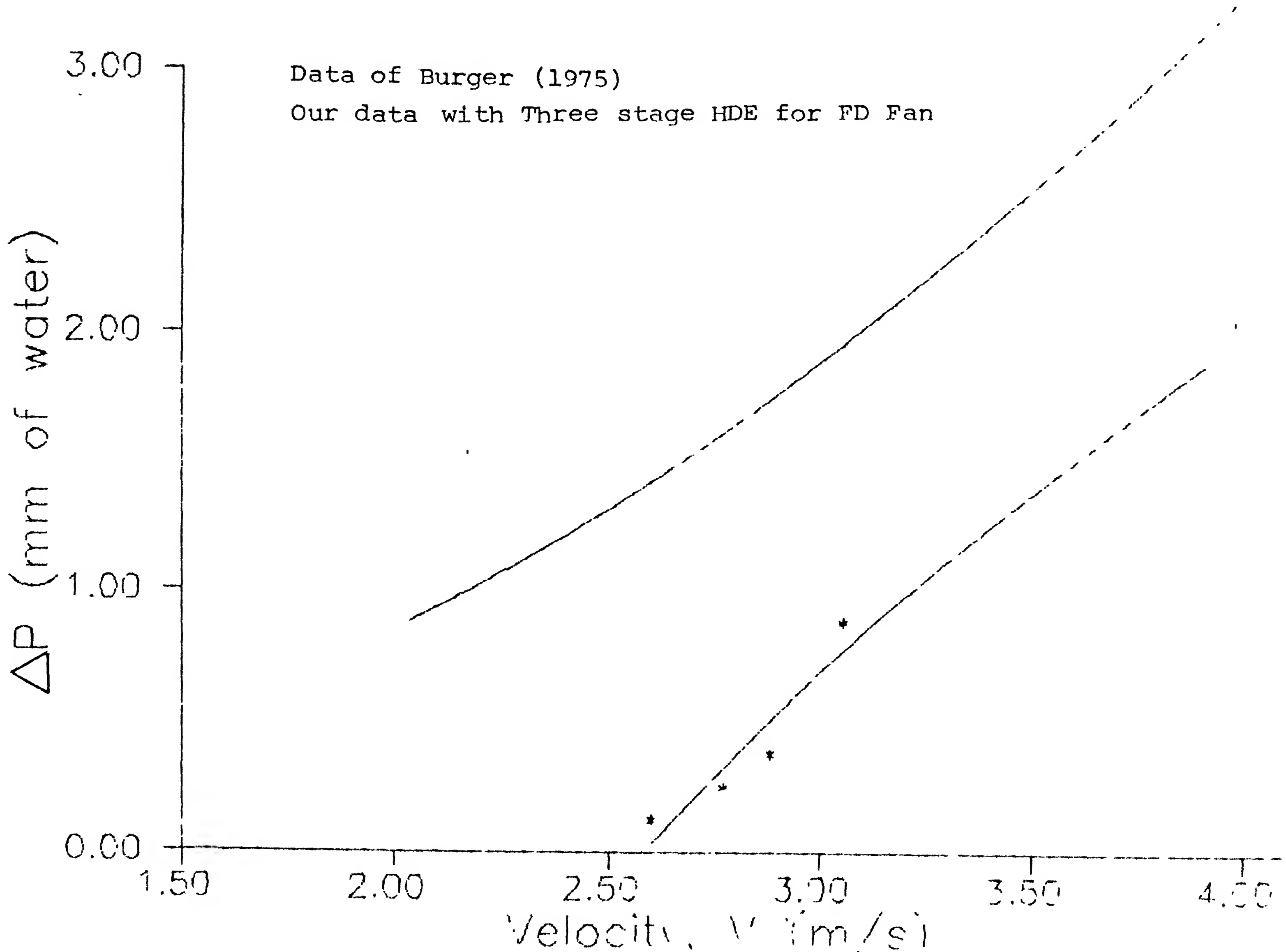


Fig: 4.21
Pressure drop Vs Velocity for cellular Drift Eliminators

drop increases with an increase in the number of stages. This is because of the larger resistance to flow. Also as the velocity increases there is an increase in the pressure drop. Within acceptable limits of drift the pressure drop from a nine-pass HDE (tested here) is lower than that of a six pass HDE or cellular drift eliminator(Burger, 1975) as shown in Figure 4.21

4.4 Optimum Angle of Orientation

Conventional type of DE can be set at any angle. As the choice is available we would like to use an optimum angle. The optimum angle of the DE plate would be a function of both the drift and pressure drop characteristics. Drift represents a loss of water (to be made through the make up water) and hence represents a cost C1. The pressure drop across the DE results in an expenditure in the form of increased power for the fan. This power loss because of the additional power of the fan is represented by a cost C2. Both C1 and C2 are functions of θ. In order to determine the optimum value of θ, the cost analysis given below can be used.

The cost of water for industrial purposes is taken to be Rs. 2.00 per 1000 litres and the cost of power as Rs. 1.00 per kWh. Costs C1 and C2 are given by :

$$C1 = 200\, M_d/1000 = 0.2\, M_d \text{ paise/h} \tag{4.1}$$

where M_d = rate of drift loss in kg/h.

Power loss due to pressure drop $p = p\,V = \rho_v g\, h\, V$.

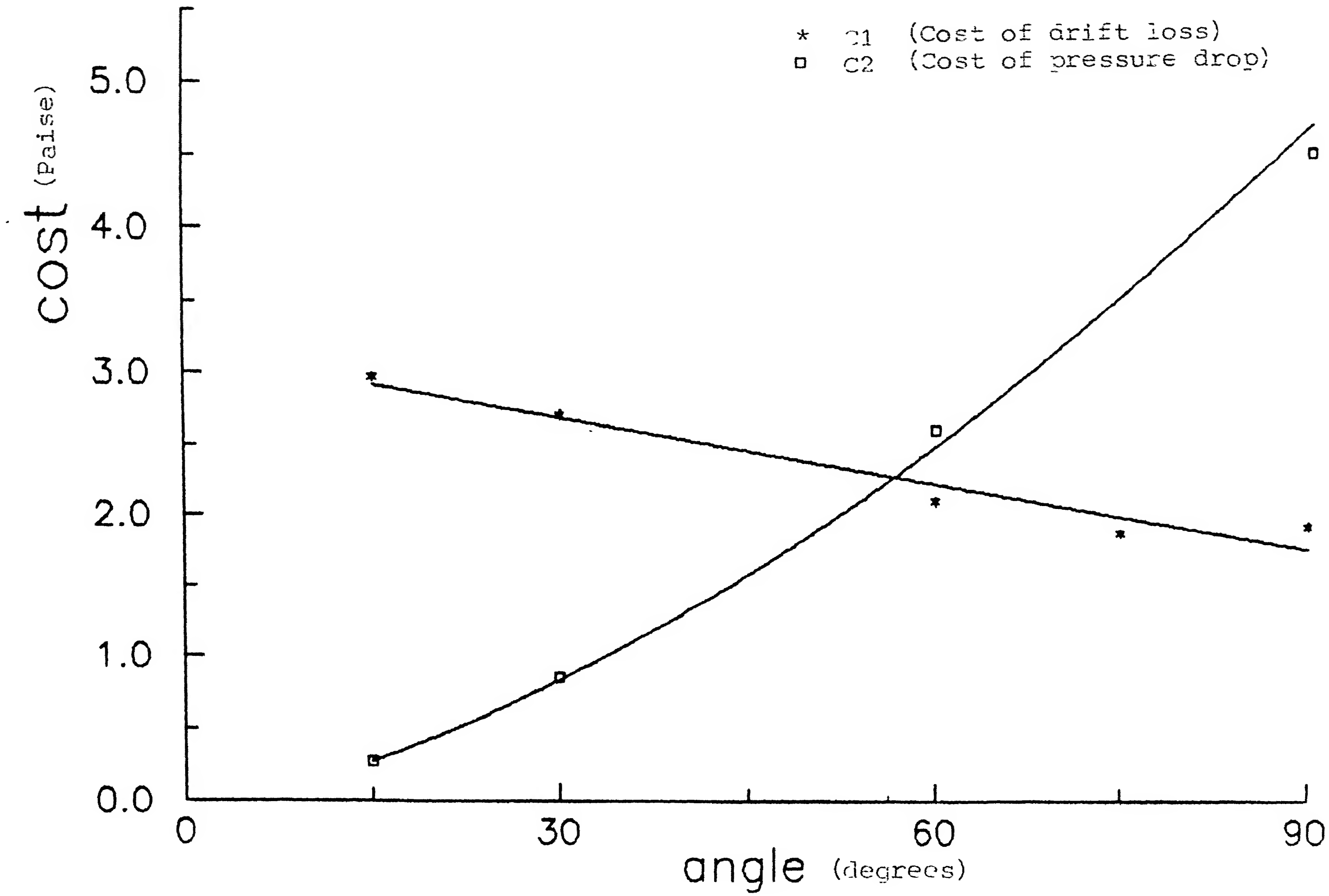


Fig: 4.22
C1,C2 Vs Inclination Angle for FD fan with 3 stage CA DE

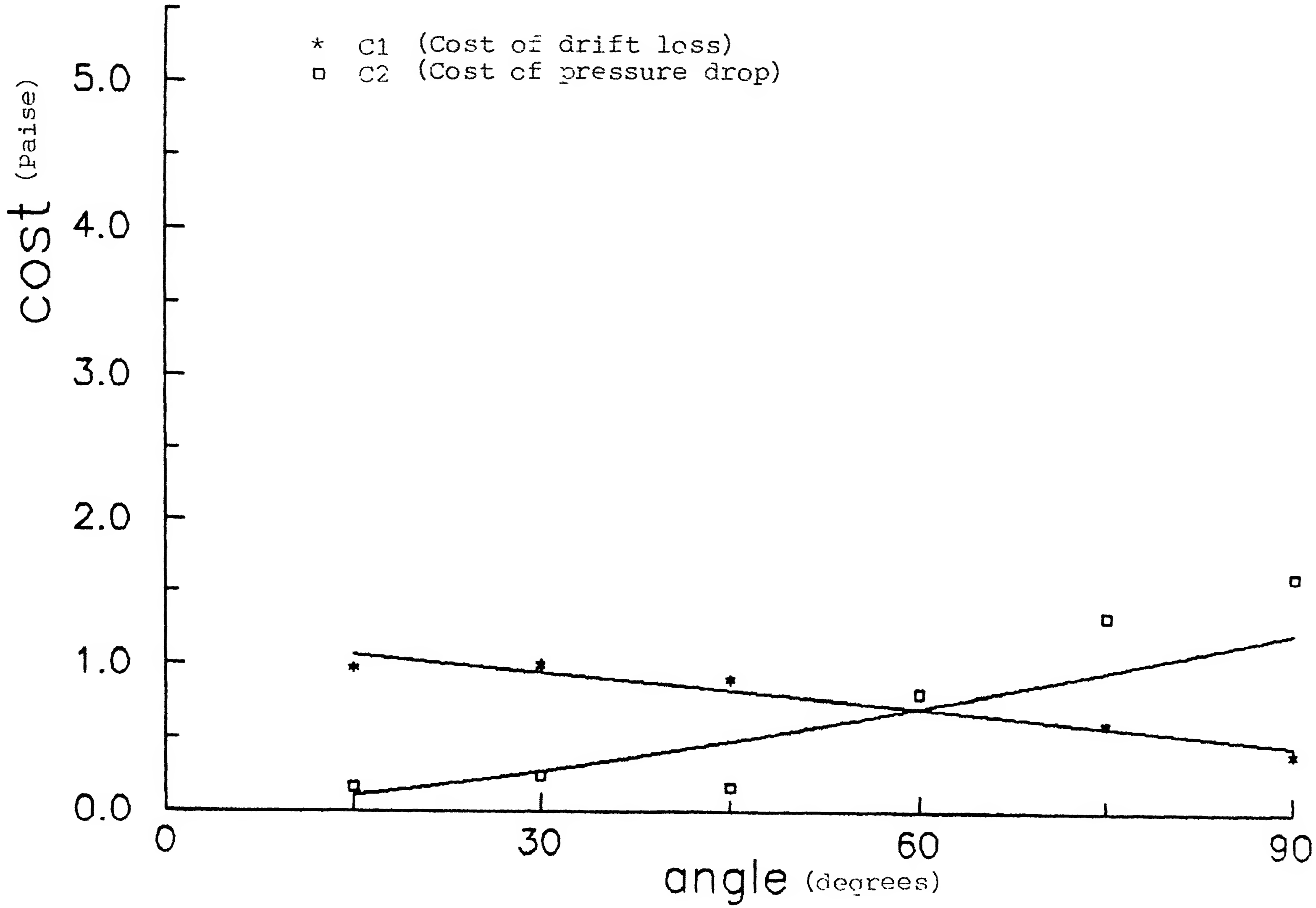


Fig: 4.23
C1,C2 Vs Inclination Angle for ID fan with 3 stage CA DE

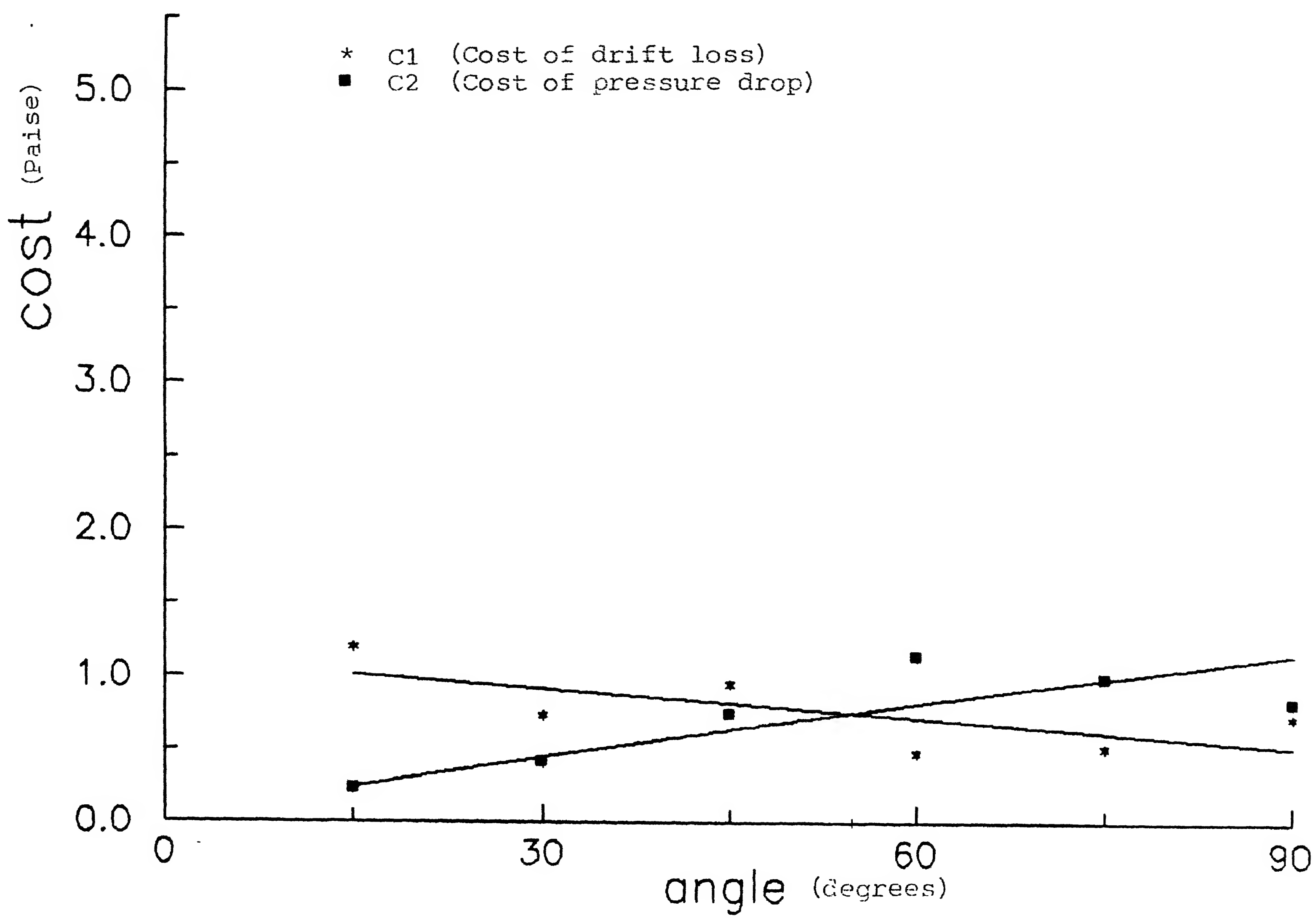


Fig: 4.24
C1,C2 Vs Inclination Angle for FD fan with 3 stage CONC DE

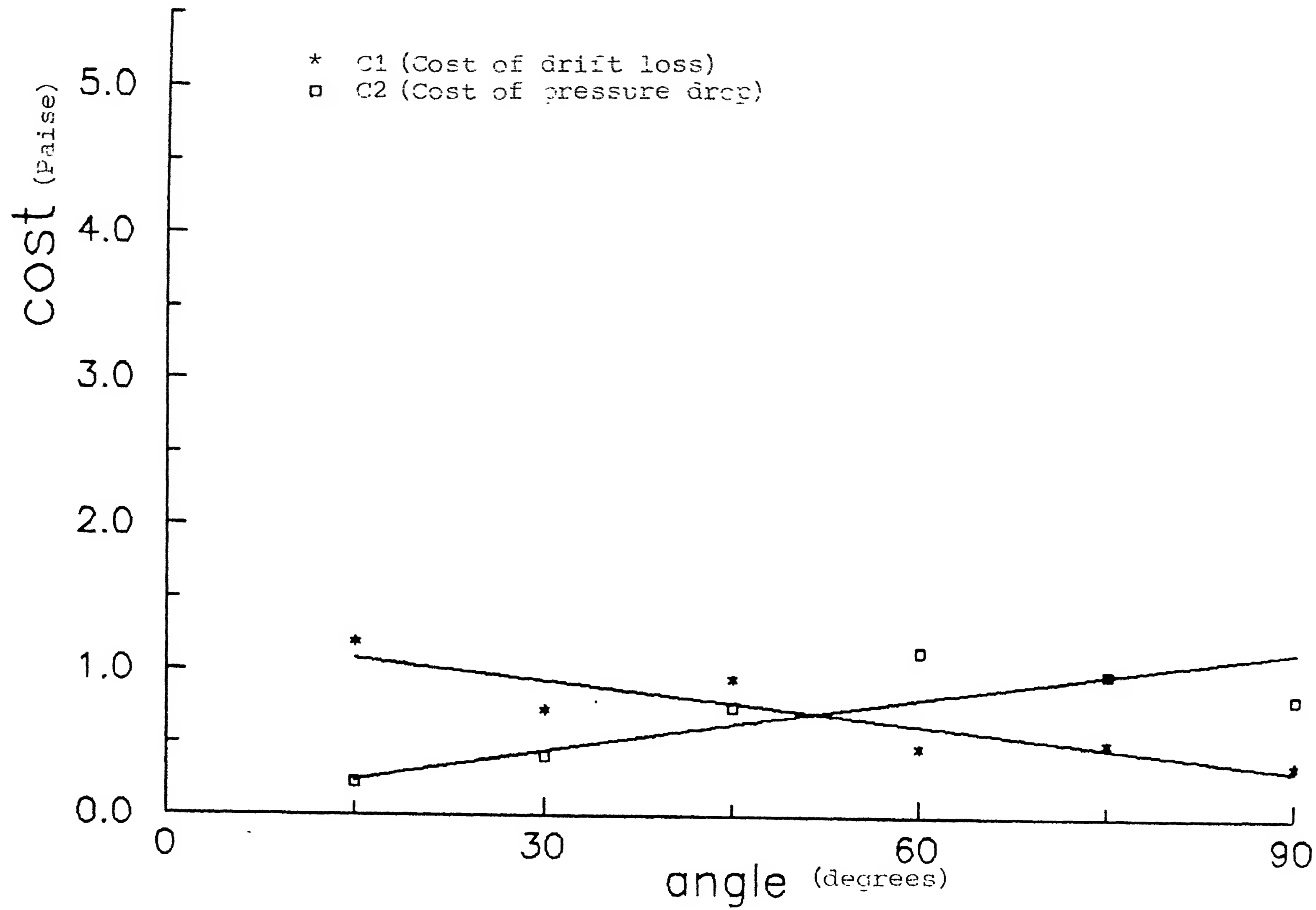


Fig: 4.25
C1,C2 Vs Inclination Angle for ID fan· with 3 stage CONC DE

Cost of this power $C2 = (\rho_v ghV)\ 100\ 10^{-3} / (3.6\ \rho_0)$

$$C2 = \frac{\rho_v ghV}{3.6} \times 10^{-7} \text{ paise/h} \qquad (4.2)$$

where V = volumetric flow rate(m^3/h),

g = acceleration due to gravity(m/s^2),

h = pressure drop in mm of water,

ρ_v = density of water kg/m^3.

Sample plots of cost versus θ are given in 4.22 and 4.23for CADE and in Figures 4.24 and 4.25 for CDE with both FD and ID fansfor 230 VAC..It can be seen from these figures that the two costs become equal for a particular value of θ which may be considered as the breakeven point or the optimum value of θ. For this value of θ, DE can become cost effective. Based upon the cost analysis presented above, the the optimum or breakeven values from the sample plots is given below:

Three stages of CADE with FD fan $\theta = 56^\circ$,

Three stage of CADE with ID fan $\theta = 59^\circ$,

Three stage of CDE with FD fan $\theta = 55^\circ$,

Three stage of CDE with ID fan $\theta = 51^\circ$.

It may be noted here that as the cost of water and power change, the optimum values of the angles would also change. The values indicated above can only be approximate values. A better method however will be to take a reasonable value of the plant life and over that period one could estimate average costs of industrial water and power and then plot C1 and C2 versus θ to get an optimum value. Both costs may not increase in the same proportion.

4.4 Coefficient Of Performance

The refrigeration system data were recorded for values of θ in the range of 15 to 90 and for supply voltage varying from 230 VAC to 160 VAC. The data are given in Tables 4.11 through 4.16 for CADE and Tables 4.17 and 4.18 for CDE. The COP was determined and was plotted versus θ, and the curves are shown in Figures 4.26 through 4.31 for CADE and Figures 4.32 and 4.33 for CDE.

From these curves it can be inferred that as θ increase, COP decreases. This may be attributed to increased resistance to flow which leads to a lower volumetric discharge and thus a lower velocity of flow past the condenser coil resulting in a lower value of the heat transfer coefficient. It is seen from Figures 4.26, 4.27 and 4.28 that for a given value of θ, the COP with a single stage of CADE is is lower than that obtained with 2 and 3 stages of CADE.

It is seen from Figures 4.26, 4.27 and 4.28 that for a given a given value θ, COP with a single stage of CADE with FD fan is slightly lower than that obtained for two and three stage of CADE. This appears to be unrealistic and may be attributed to reduced supply voltage due to voltage fluctuations and relatively high dbt and high RH of the ambient air resulting in lower COP. A similar trend is seen in Figures 4.29 (for a single stage using an ID fan) when compared with Figures 4.30 and 4.31.

An experimental investigation by Singh(1989) reported the effect of drift eliminator orientation on the COP of a refrigeration system. The COP values in the present work were

<u>TABLE 4.11</u>

Refrigeration and packing performance data for CADE

Angle of Inclination: 15°, Air inlet area: 0.1006 m^2
Make up water: 0.518 kg/min., Water flow rate: 30 kg/min.

V (Volts)	No. of stages n	Compressor Pressures P_d (bar)	P_s (bar)	Compressor Temperatures Inlet T_3	Outlet T_1	Condensers outlet T_2	Evaporator outlet T_4	Packing Temperatures Above T_5	Below T_6	Tank water Temperatures T_7
FD FAN										
230		16.62	5.03	29.1	110.6	34.44	28.30	32.78	32.78	31.11
200	1	16.96	5.17	30.0	111.7	33.33	28.89	35.00	34.72	36.39
160		17.03	5.17	30.1	109.5	30.00	29.17	36.67	35.83	33.89
230		16.75	5.06	28.00	108.5	36.50	26.90	37.22	36.67	35.00
200	2	17.17	5.17	29.20	111.7	36.11	28.10	37.78	37.22	35.56
160		17.10	5.17	29.40	110.6	32.22	28.30	38.33	37.78	35.83
230		16.55	4.96	26.40	100.6	28.10	25.00	37.78	37.22	35.56
200	3	17.10	5.17	27.20	107.5	32.20	26.10	38.61	37.50	35.33
160		17.24	5.17	27.30	109.5	33.33	26.70	38.89	38.33	35.56
ID FAN										
230		16.20	4.96	26.40	102.5	30.00	25.00	33.61	32.22	31.67
200	1	17.03	5.17	28.30	105.0	31.94	26.90	36.11	33.39	33.89
160		17.10	5.17	29.20	106.7	33.33	27.80	37.22	35.33	35.00
230		17.03	5.17	23.90	97.5	28.33	22.80	32.22	30.83	30.56
200	2	17.44	5.24	27.80	105.6	32.22	26.70	35.28	33.33	32.78
160		17.72	5.24	28.90	106.7	33.33	27.80	35.83	34.44	33.61
230		17.24	5.17	30.30	100.6	33.33	28.90	36.67	34.72	34.17
200	3	17.31	5.17	30.20	110.0	33.33	28.90	36.94	35.00	35.00
160		17.24	5.17	31.40	106.4	35.00	30.00	38.33	36.39	35.28

TABLE 4.12

Refrigeration and packing performance data for $CADE_2$
Angle of Inclination: 30°, Air inlet area: 0.1006 m^2
Make up water: 0.518 kg/min., Water flow rate: 20 kg/min.

V (Volts)	No. of stages n	Compressor Pressures P_d (bar)	P_s (bar)	Compressor Temperatures Inlet T_3	Outlet T_1	Condensers Outlet T_2	Evaporator outlet T_4	Packing Temperatures Above T_5	Below T_6	Tank water Temperatures T_7
FD FAN										
230		16.60	5.17	26.4	97.0	29.44	25.00	34.44	33.00	33.33
200	1	16.69	5.17	28.3	105.0	35.56	27.20	36.11	35.28	35.56
160		16.69	5.17	29.1	108.5	33.33	29.10	37.50	36.39	36.67
230		16.60	5.03	27.8	102.0	30.83	26.70	34.44	33.33	32.22
200	2	16.60	5.17	28.9	106.7	31.11	27.80	36.11	34.44	33.61
160		16.60	5.17	28.6	108.0	31.67	27.50	35.83	35.00	33.89
230		16.82	5.17	29.4	107.8	32.22	27.00	36.11	35.00	35.00
200	3	16.69	5.17	29.7	110.0	32.78	28.60	36.67	35.28	33.61
160		16.69	5.17	29.2	107.8	33.06	28.10	37.22	36.39	36.11
ID FAN										
230		16.55	5.10	26.4	103.3	30.56	25.00	35.00	32.78	32.22
200	1	16.96	5.17	28.3	110.0	32.78	26.90	36.11	34.72	34.17
160		17.03	5.24	28.6	110.0	33.06	27.20	37.22	35.00	34.44
230		17.10	5.17	30.0	110.6	34.17	28.90	38.33	36.39	36.11
200	2	17.03	5.17	30.0	113.0	34.72	29.40	38.89	36.67	36.11
160		17.10	5.17	30.7	115.8	36.11	30.00	39.44	37.22	36.39
230		16.62	4.96	24.2	97.3	27.50	22.80	35.00	36.39	27.50
200	3	17.10	5.17	27.0	107.8	30.00	25.60	33.61	31.94	31.11
160		17.10	5.20	28.1	110.0	31.11	26.70	35.00	33.61	31.67

TABLE 4.13

Refrigeration and packing performance data for CADE
Angle of Inclination: 45°, Air inlet area: 0.1006 m^2
Make up water: 0.518 kg/min., Water flow rate: 80 kg/min.

V (Volts)	No. of stages n	Compressor Pressures P_d (bar)	P_s (bar)	Compressor Temperatures Inlet T_3	Outlet T_1	Condensers Outlet T_2	Evaporator Outlet T_4	Packing Temperatures Above T_5	Below T_6	Tank water Temperatures T_7
FD FAN										
230		17.24	5.10	27.0	108.0	30.56	25.60	31.44	33.33	33.06
200	1	17.24	5.17	28.6	105.1	32.22	27.50	35.83	31.72	33.89
160		17.24	5.24	29.4	110.3	33.06	28.30	36.94	36.11	35.00
230		16.90	4.96	20.3	97.5	30.56	18.90	34.44	33.61	31.67
200	2	17.24	5.17	23.1	107.2	31.67	21.70	36.11	35.28	33.61
160		17.24	5.24	23.3	109.5	32.78	21.90	36.67	35.83	34.17
230		17.24	5.17	30.5	108.0	33.89	29.20	37.78	36.11	35.00
200	3	17.24	5.24	31.6	112.2	34.44	30.56	38.89	37.22	36.11
160		17.24	5.24	31.7	113.9	35.00	30.56	39.44	37.73	36.11
ID FAN										
230		16.55	5.17	28.6	106.7	33.06	27.20	37.22	35.28	35.00
200	1	17.03	5.20	29.5	111.7	33.33	28.10	37.78	35.56	35.00
160		17.10	5.31	30.3	112.5	33.85	28.90	38.33	36.11	36.39
230		17.10	5.24	28.8	112.0	31.67	27.80	36.67	34.72	35.00
200	2	17.24	5.38	30.0	113.9	33.89	28.90	38.33	36.39	36.67
160		17.24	5.24	31.2	117.2	35.00	30.60	39.17	36.67	37.22
230		17.24	5.17	25.3	102.2	29.44	23.90	33.61	31.67	32.22
200	3	17.78	5.52	29.1	113.0	31.67	27.80	36.39	35.00	35.56
160		17.24	6.17	25.2	103.6	28.89	23.90	33.89	31.11	30.83

TABLE 4.14

Refrigeration and packing performance data for $CADE_2$
Angle of Inclination: 60°, Air inlet area: 0.100[illegible] m^2
Make up water: 0.518 kg/min., Water flow rate: [illegible] kg/min.

V (Volts)	No. of stages n	Compressor Pressures P_d (bar)	P_s (bar)	Compressor Temperatures Inlet T_3	Outlet T_1	Condensers Cutlet T_2	Evaporator Outlet T_4	Packing Temperatures Above T_5	Below T_6	Tank Water Temperatures T_7
	FD FAN									
230		17.03	5.14	25.3	105.0	28.33	23.9	33.33	31.67	30.56
200	1	17.24	5.24	28.9	111.1	32.22	27.8	36.11	35.56	35.00
160		17.24	5.31	28.9	112.2	33.33	27.8	37.22	36.11	36.11
230		17.24	5.17	29.4	110.0	33.33	28.3	37.22	35.33	33.89
200	2	17.38	5.24	29.4	98.6	33.33	28.3	37.50	36.11	35.00
160		17.51	5.38	31.7	115.6	34.44	30.6	39.17	37.78	36.67
230		17.31	5.20	29.5	110.6	33.33	28.1	37.22	35.56	33.85
200	3	17.24	5.24	29.4	115.0	33.06	28.3	36.67	35.33	33.85
160		17.24	5.31	30.0	115.0	33.89	28.9	38.33	36.67	36.11
	ID FAN									
230		17.31	5.17	27.0	108.4	30.56	25.6	34.72	33.06	33.33
200	1	17.44	5.20	28.6	112.2	32.22	27.2	35.83	34.17	34.44
160		17.58	5.31	30.0	114.4	33.06	28.6	37.50	35.83	35.83
230		17.10	5.13	25.5	105.6	30.00	24.4	33.33	31.67	31.67
200	2	17.24	5.17	28.8	106.1	32.78	27.8	36.39	34.72	33.61
160		17.24	5.17	30.0	115.6	34.44	28.9	38.89	36.39	36.11
230		17.20	5.06	25.8	104.5	30.28	24.4	32.78	31.39	31.11
200	3	17.38	5.20	27.5	111.1	32.22	26.1	35.33	34.17	33.89
160		17.24	5.31	28.6	110.0	33.06	27.2	36.94	35.28	35.00

TABLE 4.15

Refrigeration and packing performance data for CASE 2
Angle of Inclination: 75°, Air inlet area: 0.1906 m^2
Make up water: 0.518 kg/min., Water flow rate: [illegible] kg/min.

V (Volts)	No. of stages n	Compressor Pressures P_d (bar)	P_s (bar)	Compressor Temperatures Inlet T_3	Outlet T_1	Condensers Outlet T_2	Evaporator Outlet T_4	Packing Temperatures Above T_5	Below T_6	Tank water Temperatures T_7
FD FAN										
230	1	16.55	5.03	25.8	98.6	29.44	24.4	33.06	31.94	32.78
200		16.75	5.17	26.7	107.2	32.22	25.6	34.72	33.[illegible]	34.17
160		16.96	5.17	27.8	107.8	31.39	26.7	35.00	34.17	34.44
230	2	16.75	5.13	28.6	108.9	32.22	27.5	35.11	34.72	35.56
200		16.96	5.13	28.6	111.7	33.06	27.5	36.39	35.56	36.11
160		17.10	5.17	29.4	111.1	33.61	28.3	37.50	36.39	[illegible].67
230	3	16.96	4.96	27.8	108.0	31.67	26.7	35.56	34.44	34.72
200		16.75	5.07	28.3	112.3	36.11	27.2	36.39	35.28	35.56
160		16.96	5.17	28.9	113.3	33.06	27.8	37.22	35.33	35.83
ID FAN										
230	1	17.38	5.17	26.4	102.8	30.28	25.0	34.44	33.06	31.67
200		17.72	5.38	28.6	107.2	32.78	27.2	36.11	34.44	34.17
160		17.86	5.38	29.5	109.5	33.89	28.1	37.50	36.11	35.83
230	2	17.38	4.96	25.4	105.0	30.56	24.4	33.89	32.78	37.22
200		17.79	5.10	27.2	112.2	31.39	26.1	35.56	34.17	33.61
160		17.93	5.13	30.0	116.1	33.89	28.1	37.78	36.39	35.28
230	3	17.24	5.17	25.3	101.1	23.90	23.8	31.67	30.83	29.44
200		17.72	5.38	28.1	109.5	26.70	31.6	35.28	34.17	32.22
160		17.72	5.31	28.6	111.1	27.20	32.5	35.56	34.44	32.78

TABLE 4.16

Refrigeration and packing performance data for CNDE.
Angle of Inclination: 90°, Air inlet area: 0.1006 m
Make up water: 0.513 kg/min., Water flow rate: [illegible] kg/min.

V (Volts)	No. of stages n	Compressor Pressures P_d (bar)	P_s (bar)	Compressor Temperatures Inlet T_3	Outlet T_1	Condensers Outlet T_2	Evaporator Outlet T_4	Packing Temperatures Above T_5	Below T_6	Tank water Temperatures T_7
FD FAN										
230		17.24	5.00	30.2	100.3	33.33	29.40	38.61	36.11	32.22
200	1	17.24	5.10	27.8	98.9	31.11	26.70	39.44	39.33	31.11
160		17.24	5.17	27.8	102.2	30.56	26.70	40.00	38.39	32.78
230		16.41	4.83	27.5	88.3	28.89	26.39	32.78	31.11	30.50
200	2	16.55	5.03	27.2	97.0	28.89	26.10	33.89	33.06	30.56
160		17.03	5.24	28.6	103.9	31.67	27.50	36.11	34.72	33.06
230		16.75	5.10	25.3	92.2	29.44	23.90	36.67	35.56	31.39
200	3	16.69	5.13	27.2	103.6	30.56	26.10	38.06	35.56	33.33
160		16.75	5.17	28.6	104.5	33.33	27.50	36.11	37.22	34.44
ID FAN										
230		17.38	5.35	31.3	112.2	35.00	29.70	32.22	30.56	36.67
200	1	17.58	5.31	31.4	117.8	35.83	30.00	38.06	36.39	37.78
160		17.65	5.35	32.0	117.8	35.56	30.60	33.33	32.22	38.33
230		17.31	5.17	25.0	103.3	28.06	23.90	33.33	31.11	30.83
200	2	17.51	5.35	26.7	110.0	31.11	25.60	33.89	31.11	33.33
160		17.79	5.38	28.4	110.6	31.94	27.20	36.11	34.44	34.72
230		17.75	5.35	28.6	111.7	32.78	27.50	33.89	32.78	36.11
200	3	17.93	5.52	30.6	112.8	33.89	29.20	35.00	33.89	36.11
160		17.86	5.38	30.2	112.8	34.44	28.90	36.39	33.56	37.50

TABLE 4.17

Refrigeration and acking performance data for FD Fan with CDE. Air inlet area: 0.1006 m^2
Make up water: 0.518 kg/min., Water flow rate: 30 kg/min.

V (Volts)	Angle θ (degrees)	Compressor Pressures P_d (bar)	P_s (bar)	Compressor Temperatures Inlet T_3	Outlet T_1	Condensers Outlet T_2	Evaporator Outlet T_4	Packing Temperatures Above T_5	Below T_6	Tank Water Temperatures T_7
230		16.06	5.10	33.06	104.50	33.89	32.00	33.89	31.67	31.67
200	15	16.69	5.17	28.60	104.50	35.56	27.50	35.56	33.89	33.33
160		16.55	4.96	23.33	104.70	36.67	27.22	36.67	34.72	33.89
230		16.06	4.96	26.94	99.45	28.39	25.83	33.61	31.67	30.56
200	30	16.55	5.10	28.33	104.70	30.28	27.22	33.61	31.11	31.67
160		16.55	5.17	28.33	105.10	31.11	27.22	36.39	34.44	33.33
230		16.20	4.96	28.00	107.78	31.11	26.94	35.83	33.61	33.61
200	45	16.06	5.17	28.90	110.55	32.50	27.78	36.67	34.72	33.89
160		16.20	5.10	29.20	111.67	33.33	28.06	37.22	35.00	35.83
230		16.00	4.83	23.60	95.00	26.67	22.50	30.00	30.17	29.89
200	60	16.34	5.10	25.60	104.45	29.17	24.44	33.33	31.39	30.83
160		16.48	5.03	26.40	104.45	30.56	25.28	35.28	33.33	33.06
230		16.20	4.83	22.78	98.33	25.56	21.67	37.22	26.67	27.78
200	75	16.69	5.10	26.11	108.90	28.89	25.00	25.33	31.35	31.67
160		16.55	5.17	26.70	109.00	30.00	25.56	33.33	32.22	33.33
230		14.69	4.55	22.22	90.80	26.67	21.11	26.67	25.00	26.11
200	90	15.10	4.69	23.60	96.10	29.44	22.50	29.44	25.94	27.78
160		14.76	4.65	24.70	95.60	29.44	23.33	29.44	27.50	28.33

TABLE 4.18

Refrigeration and Packing Performance data
for ID Fan with CDE. Air inlet area: 0.1006 m²
Make up water: 0.518 kg/min., Water flow rate: [illegible] kg/min.

V (Volts)	Angle θ	Compressor Pressures P_d	P_s	Compressor Temperatures Inlet	Outlet	Condensers Outlet	Evaporator Outlet	Packing Temperatures Above	Below	Tank Water Temperatures
	(degrees)	(bar)	(bar)	T_3	T_1	T_2	T_4	T_5	T_6	T_7
230	15	16.55	4.83	22.50	98.90	27.22	21.39	30.28	27.78	29.17
200		15.86	4.96	26.11	103.90	30.83	25.00	35.00	32.22	33.05
160		15.86	4.96	26.11	102.00	30.28	25.00	35.83	32.06	33.89
230	30	16.07	4.83	23.10	96.11	26.67	21.94	30.00	27.33	27.22
200		16.55	4.96	22.22	102.78	28.06	21.11	32.22	30.28	29.41
160		16.55	5.00	25.28	102.50	28.89	24.17	33.61	31.67	30.56
230	45	16.27	4.96	23.61	96.67	27.22	22.50	31.67	28.72	29.17
200		16.65	5.13	24.72	100.56	28.06	23.61	33.06	30.83	30.28
160		16.69	5.13	25.55	100.56	28.89	24.44	33.06	31.11	30.55
230	60	16.41	5.13	26.67	100.00	30.83	25.56	35.56	33.06	33.06
200		16.41	5.13	26.67	102.76	31.11	25.56	36.11	34.17	33.89
160		16.41	5.13	27.22	101.17	31.94	26.11	37.22	31.44	31.44
230	75	15.86	4.96	23.33	97.78	25.56	22.22	30.56	29.99	30.00
200		16.55	5.10	24.72	102.00	29.44	23.61	32.78	30.56	31.11
160		16.55	5.10	25.35	102.00	28.89	24.44	34.44	32.22	33.05
230	90	15.72	4.83	22.22	96.40	27.22	21.11	31.39	28.17	30.56
200		15.72	4.83	23.61	100.60	27.50	22.50	33.06	31.11	31.94
160		15.86	4.83	24.44	101.40	28.61	23.33	33.33	31.39	32.50

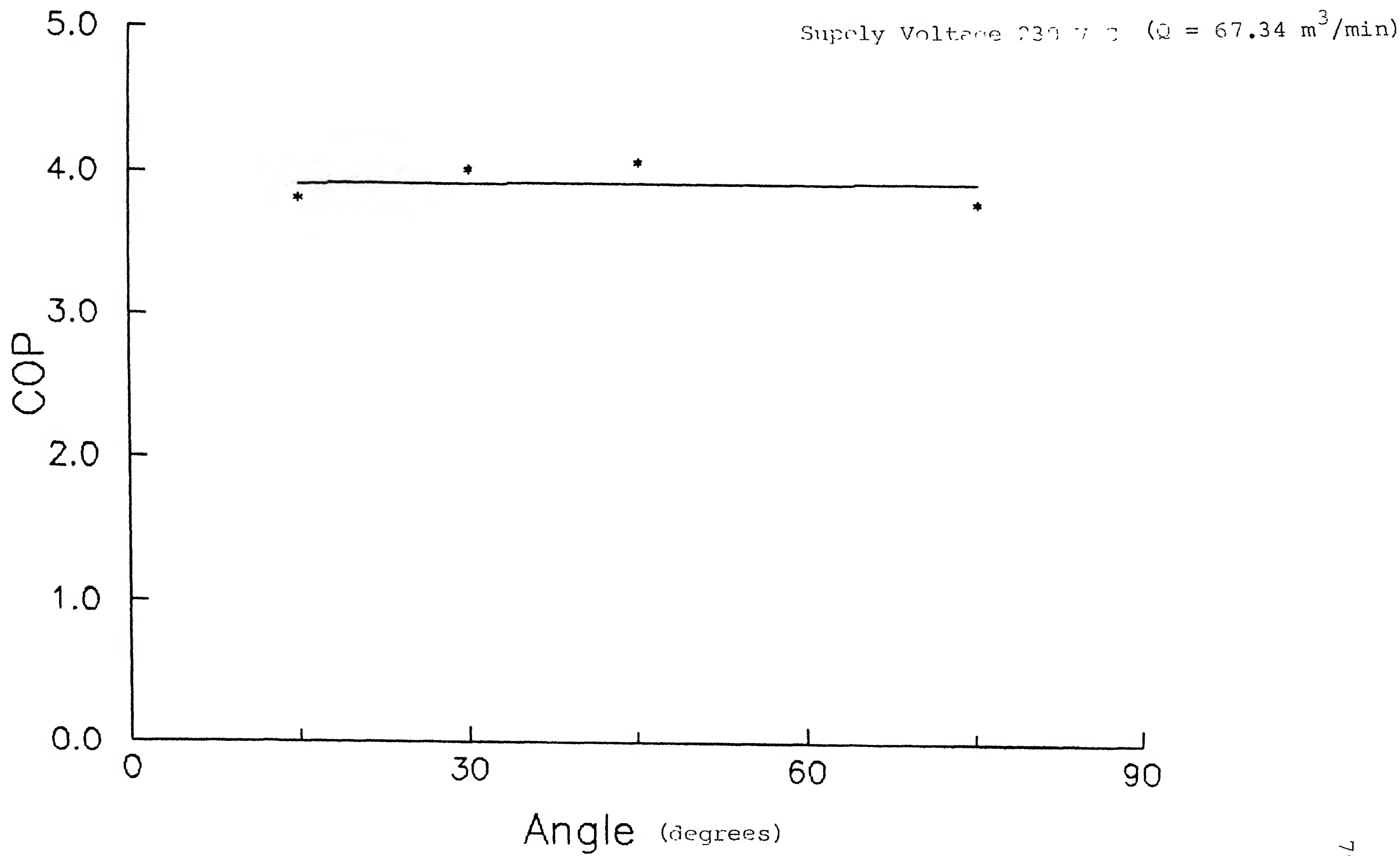


Fig: 4.26
COP Vs Inclination angle for FD fan with single stage CA DE

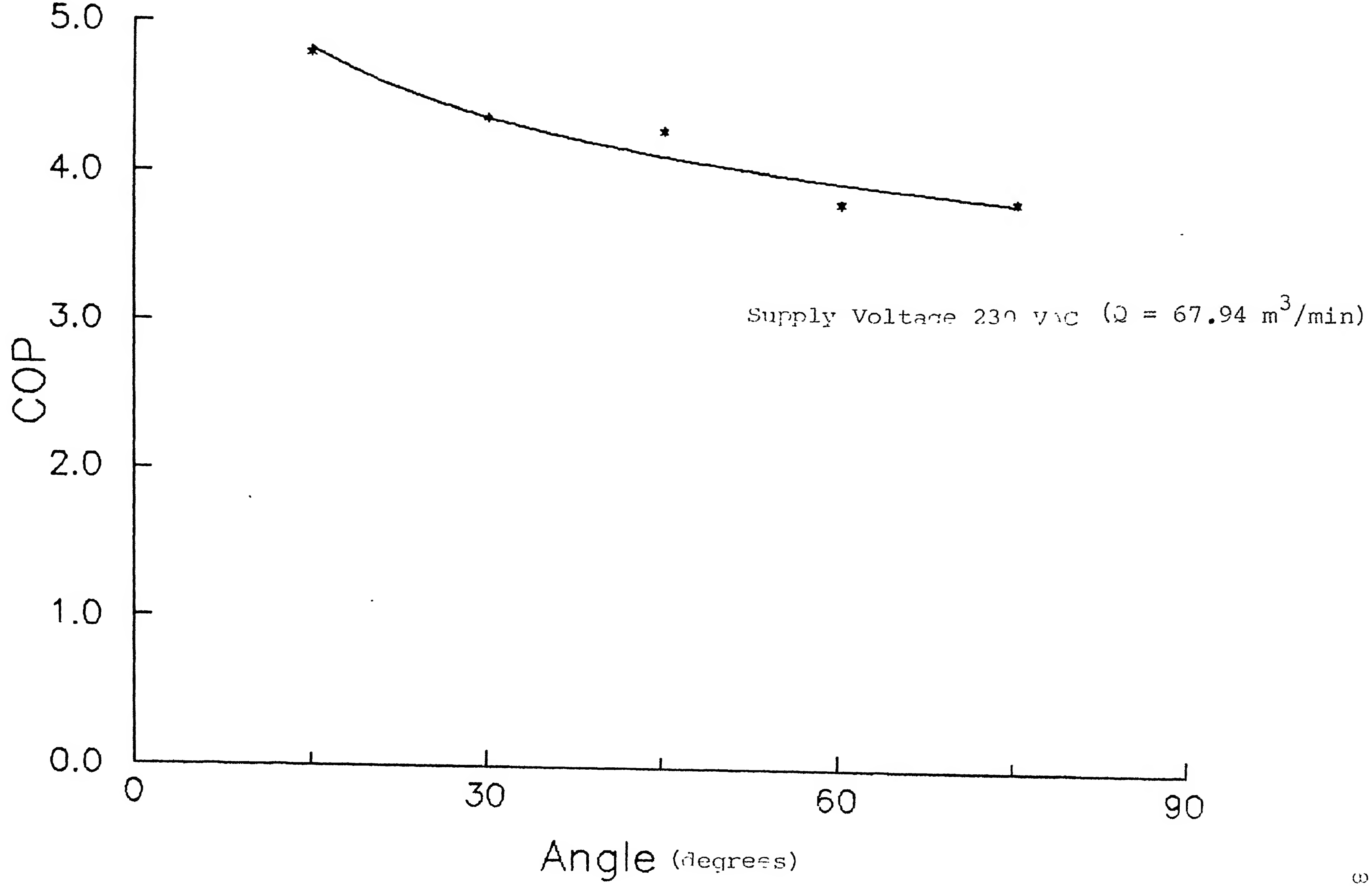


Fig: 4.27
COP Vs Inclination angle for FD fan with two stage CA DE

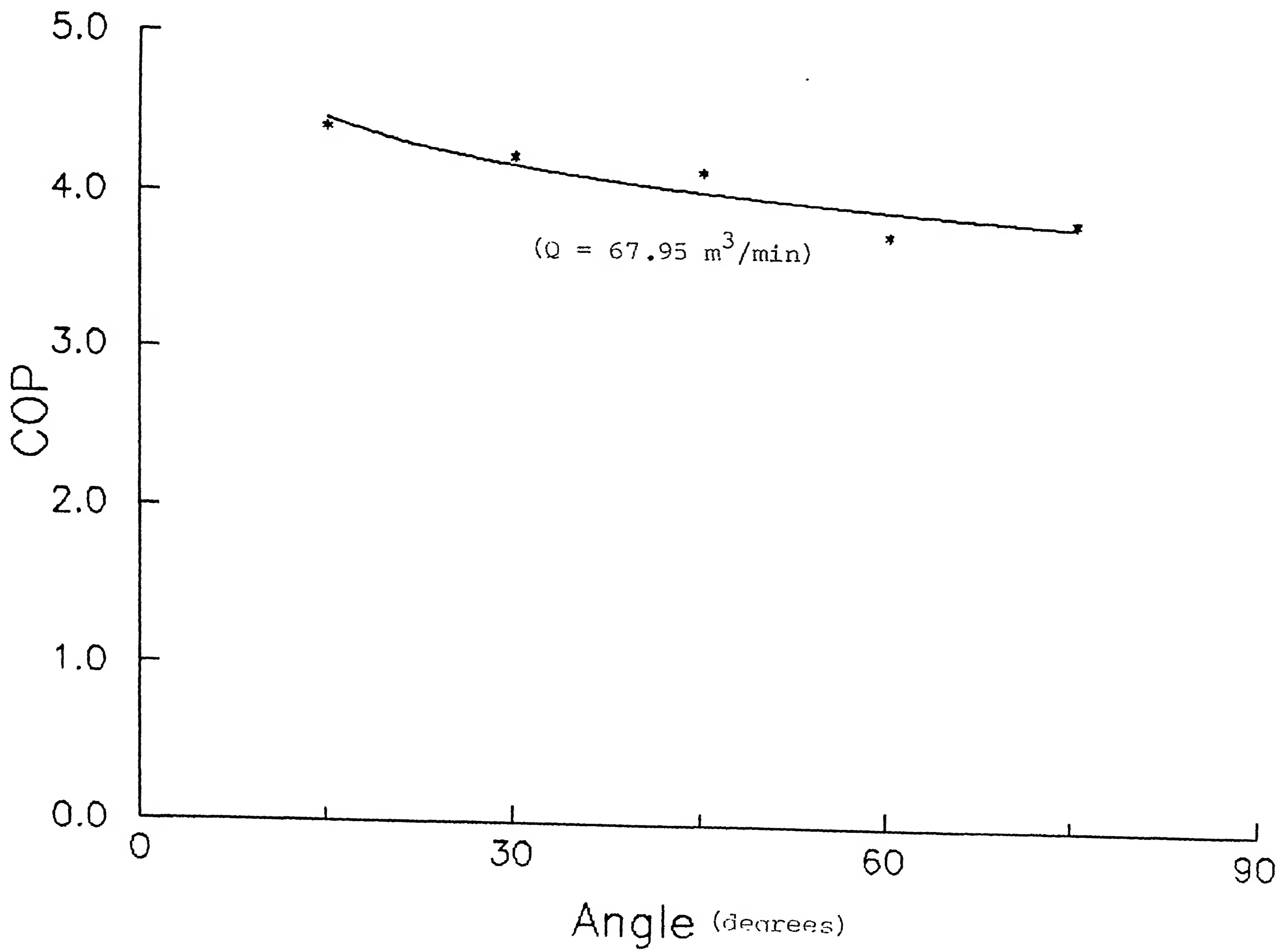


Fig: 4.25
COP Vs Inclination angle for FD fan with three stage CA DE

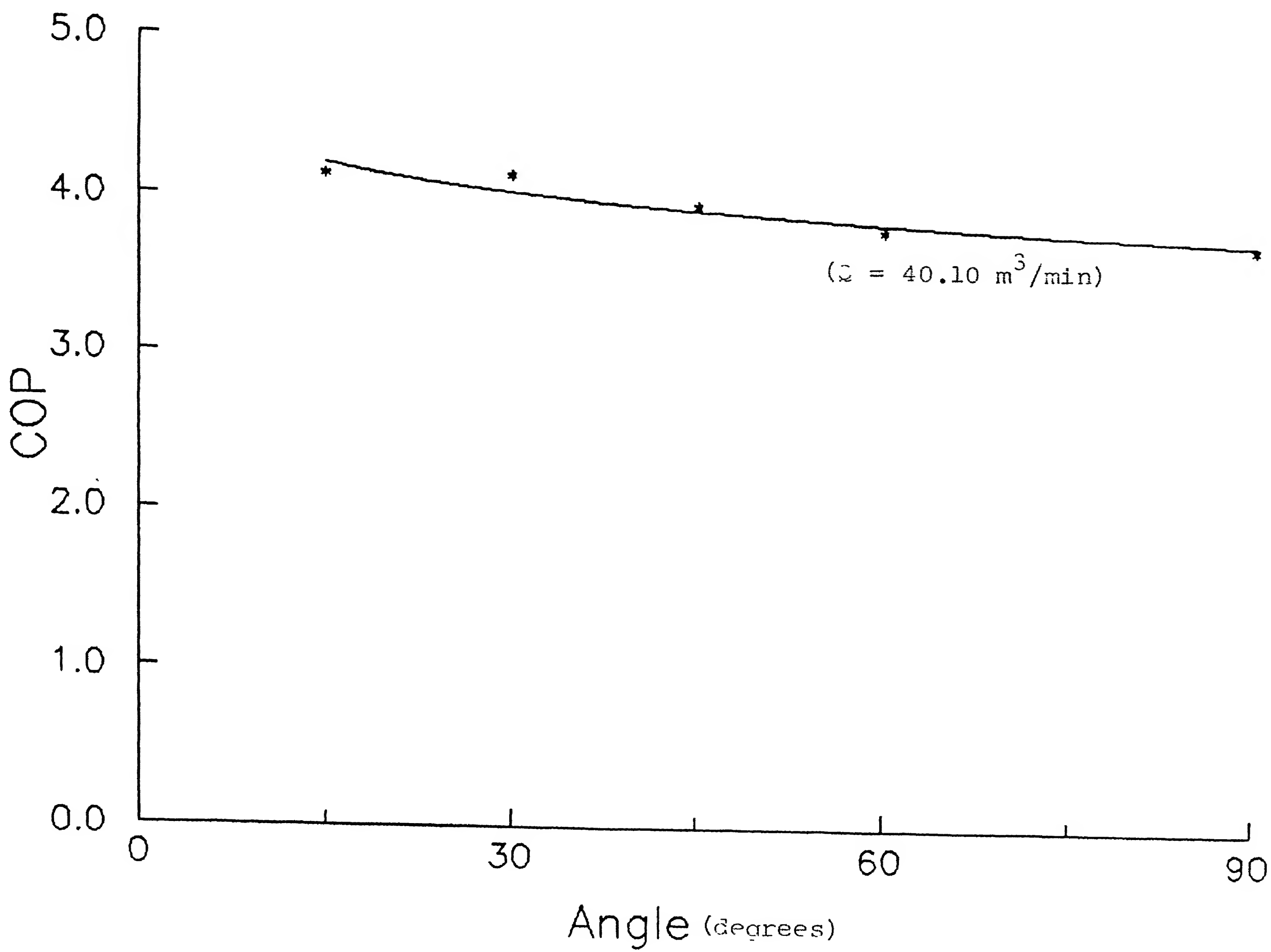


Fig: 4.2 ,
COP Vs Inclination angle for ID fan with single stage CA DE

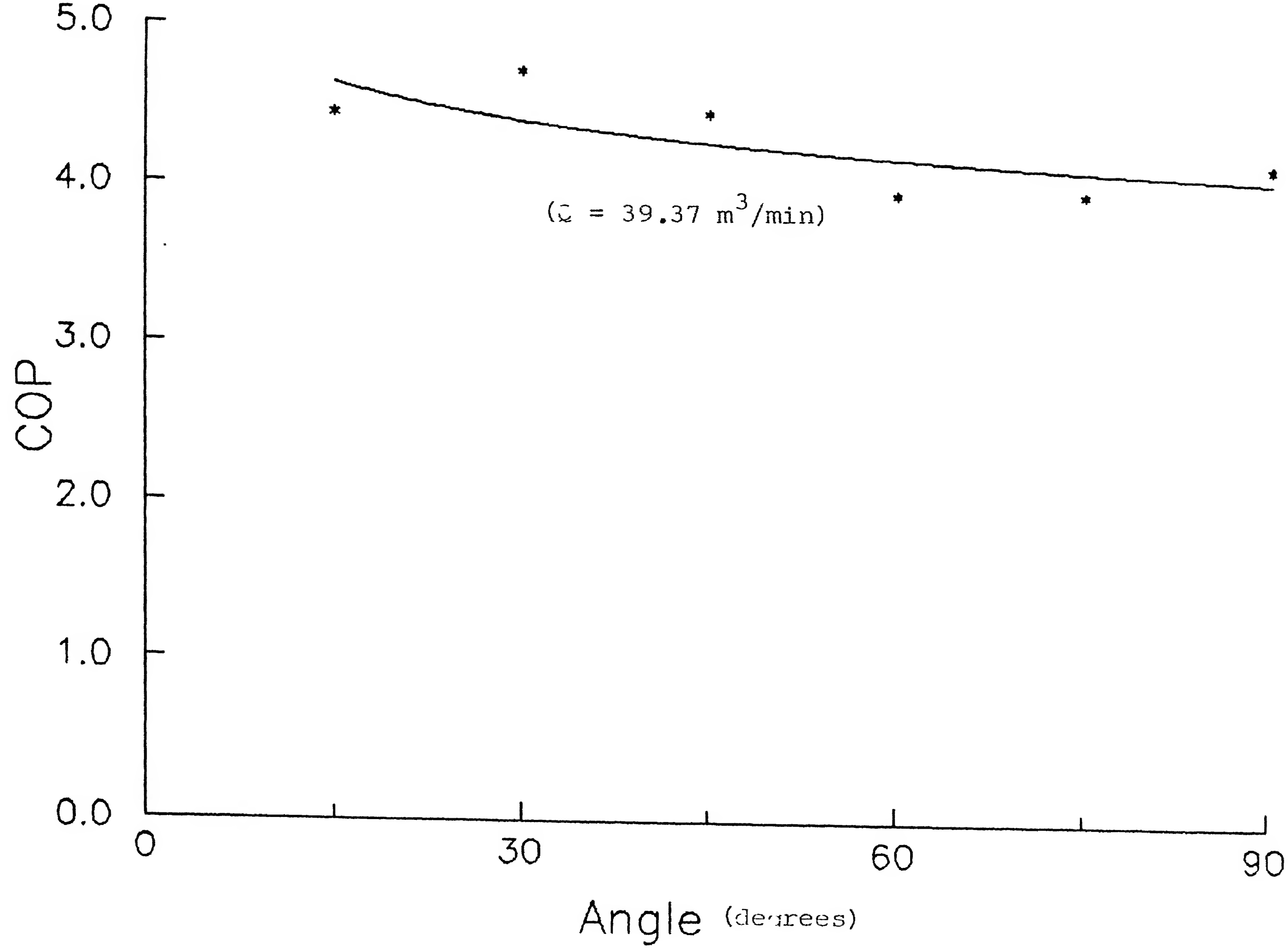


Fig: 4.30
COP Vs Inclination angle for ID fan with two stage CA DE

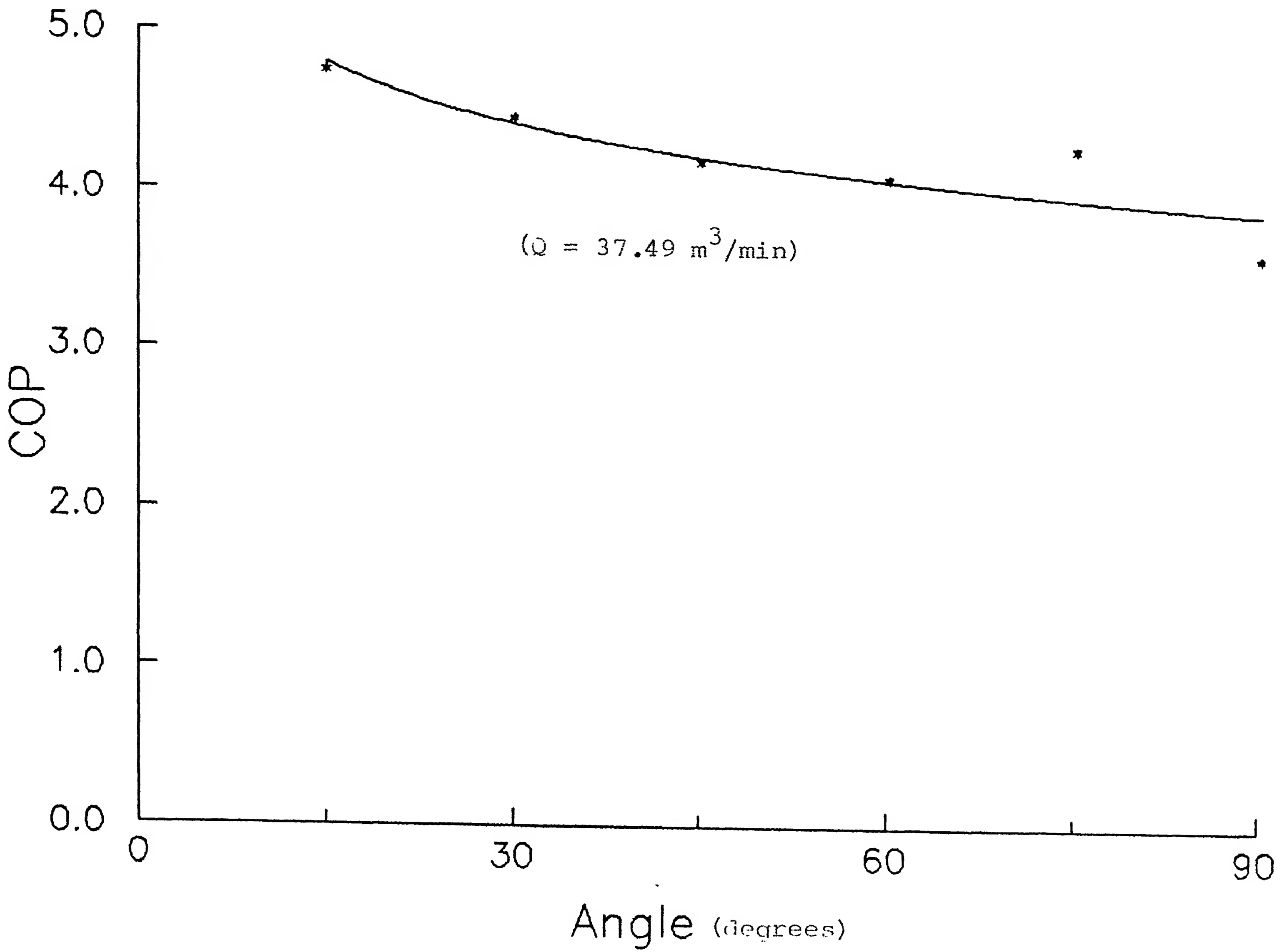


Fig: 4.31
COP Vs Inclination angle for ID fan with three stage CA DE

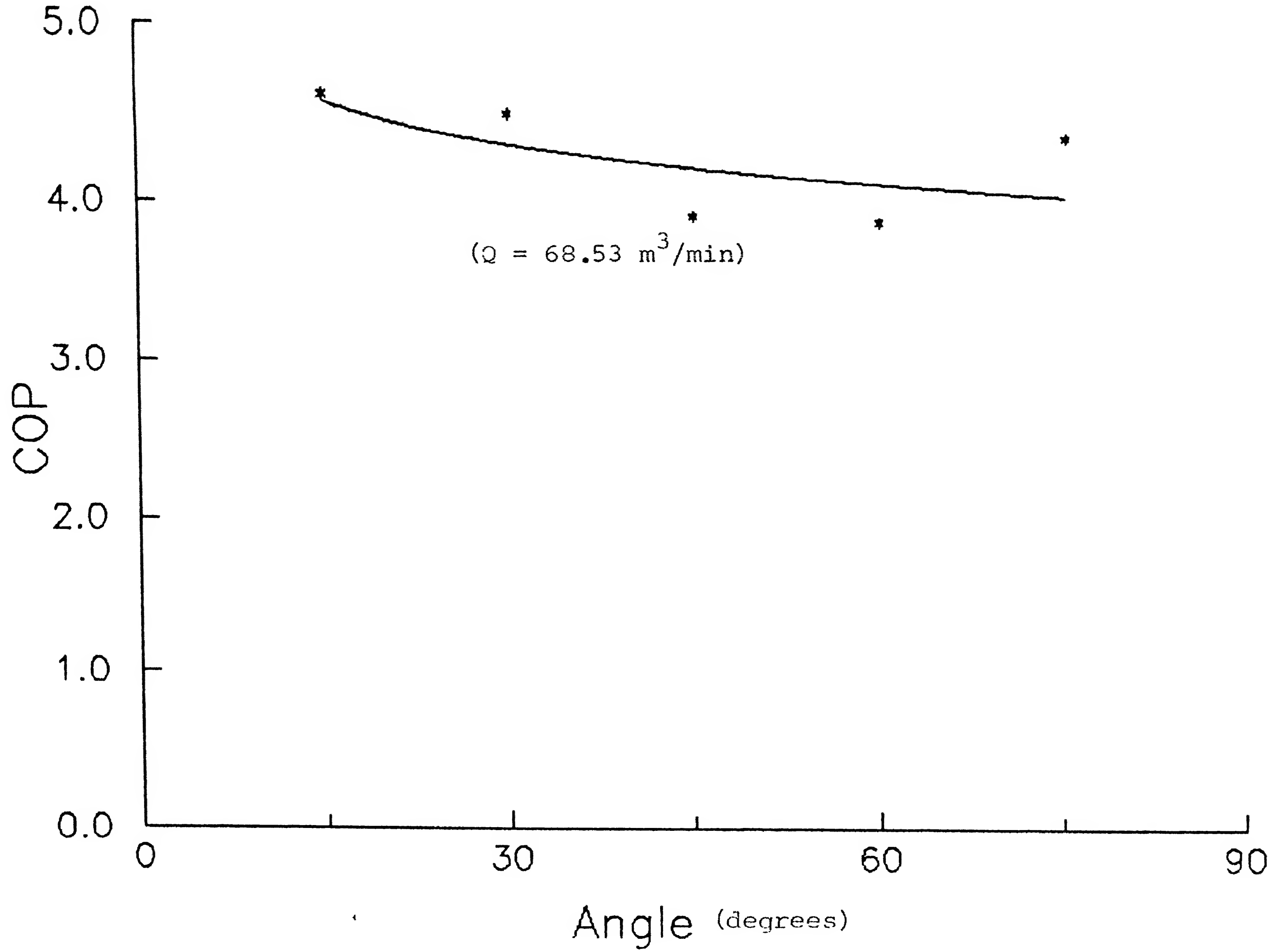


Fig: 4.32
COP Vs Inclination angle for FD fan with three stage CONC DE

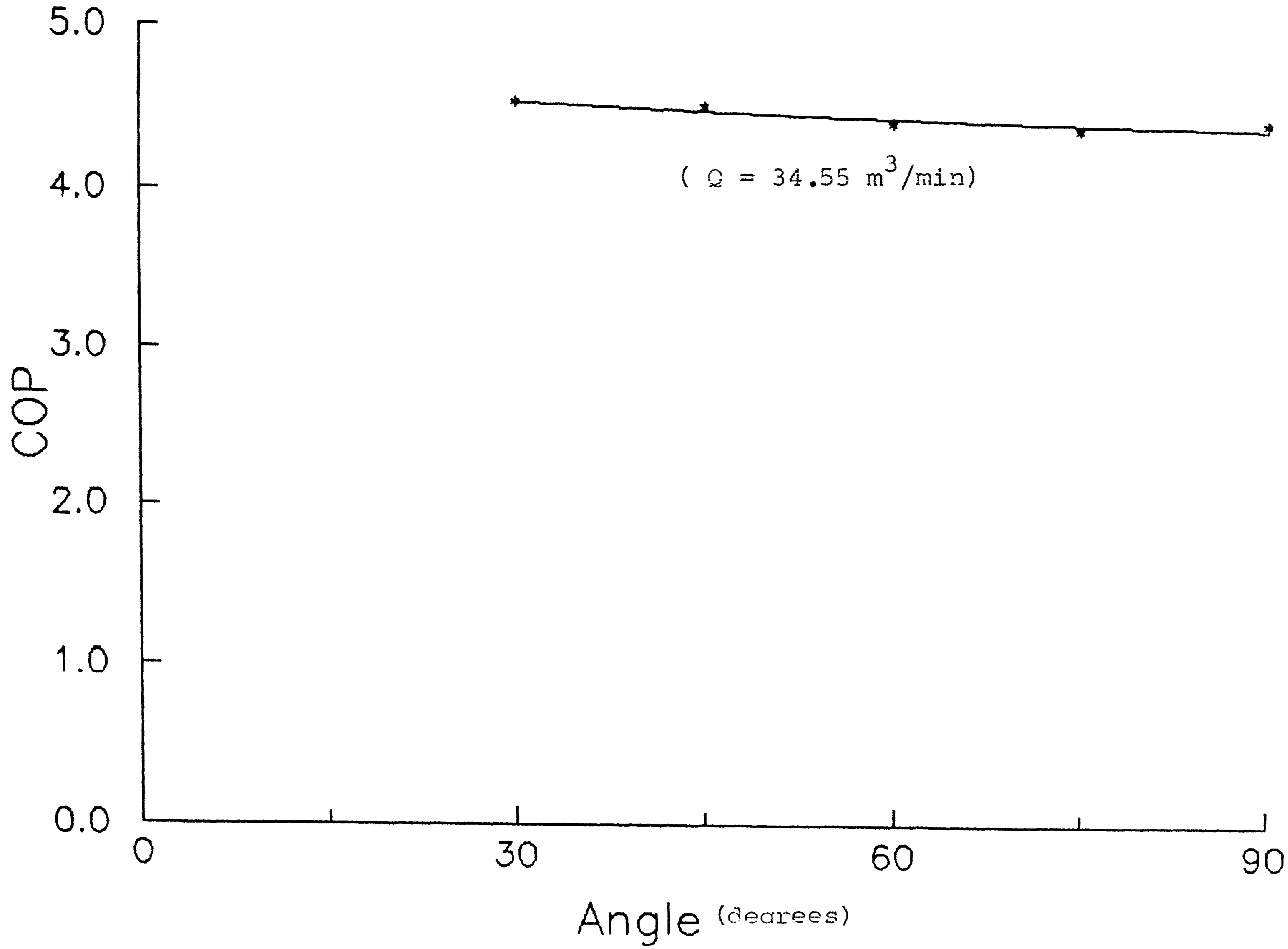


Fig: 4.33
COP Vs Inclination angle for ID fan with three stage CONC DE

TABLE 4.19

Pressure Drop Data for the Packing
Inlet Area: 0.1006 m^2
Water flow rate: 60.15 kg/min.

No. of stages						
	Δp	1.524	1.27	0.762	0.508	0.254
3	Velocity v (m/s)	13.030	11.50	11.170	10.630	10.120
2	Pressure drop Δp (mm)	10.508	0.381	0.254	0.381	0.127
	Velocity v (m/s)	12.730	12.220	11.640	11.470	[illegible].470
1	Δp Pressure drop (mm)	0.381	0.254	0.127	-	-
	Velocity v (m/s)	12.980	12.000	10.220	9.780	-

TABLE 4.20

Data for the Packing Performance Characteristic
Packing height Z = 0.105 m, Area of packing = 0.453 m^2,
Air inlet area: 0.1006 m^2

Water flow rate	Velocity v	Temperature above the packing °C	Temperature below the packing °C	Tank Temperature °C	dbt inlet °C	Ø inlet %
80.25	5.67	27.50	23.61	23.33	23.33	70
70.25	5.60	30.28	26.67	28.33	23.00	70
50.20	5.87	30.83	27.50	27.22	24.00	64
80.25	11.40	29.44	25.28	25.00	23.61	62
50.20	11.20	31.39	27.50	27.50	23.61	77
60.15	5.73	31.11	27.78	29.17	21.94	73
54.50	5.73	32.50	28.06	28.61	22.22	73

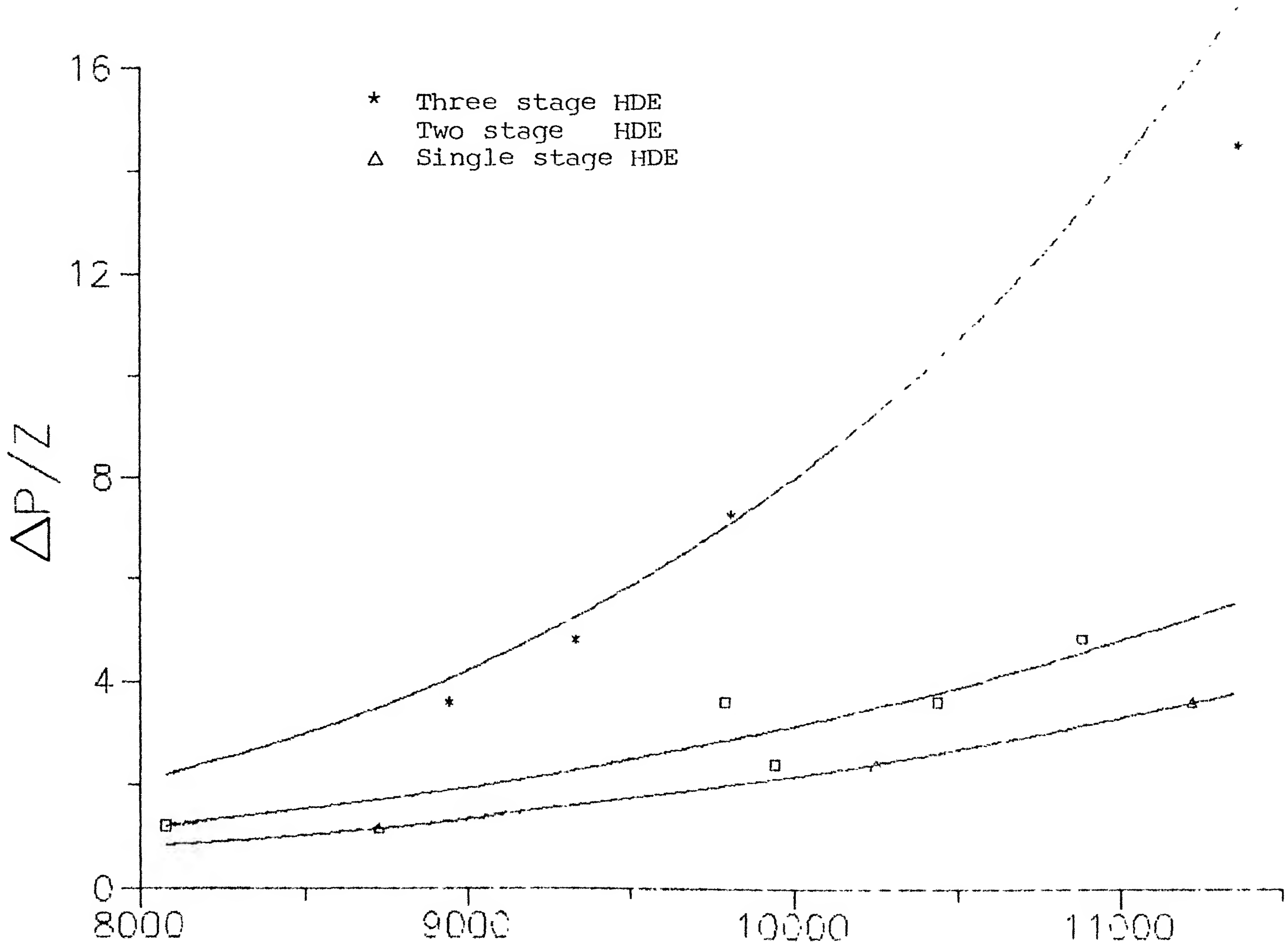


Fig: 4.34
ΔP/Z vs G for the Cellular Packing

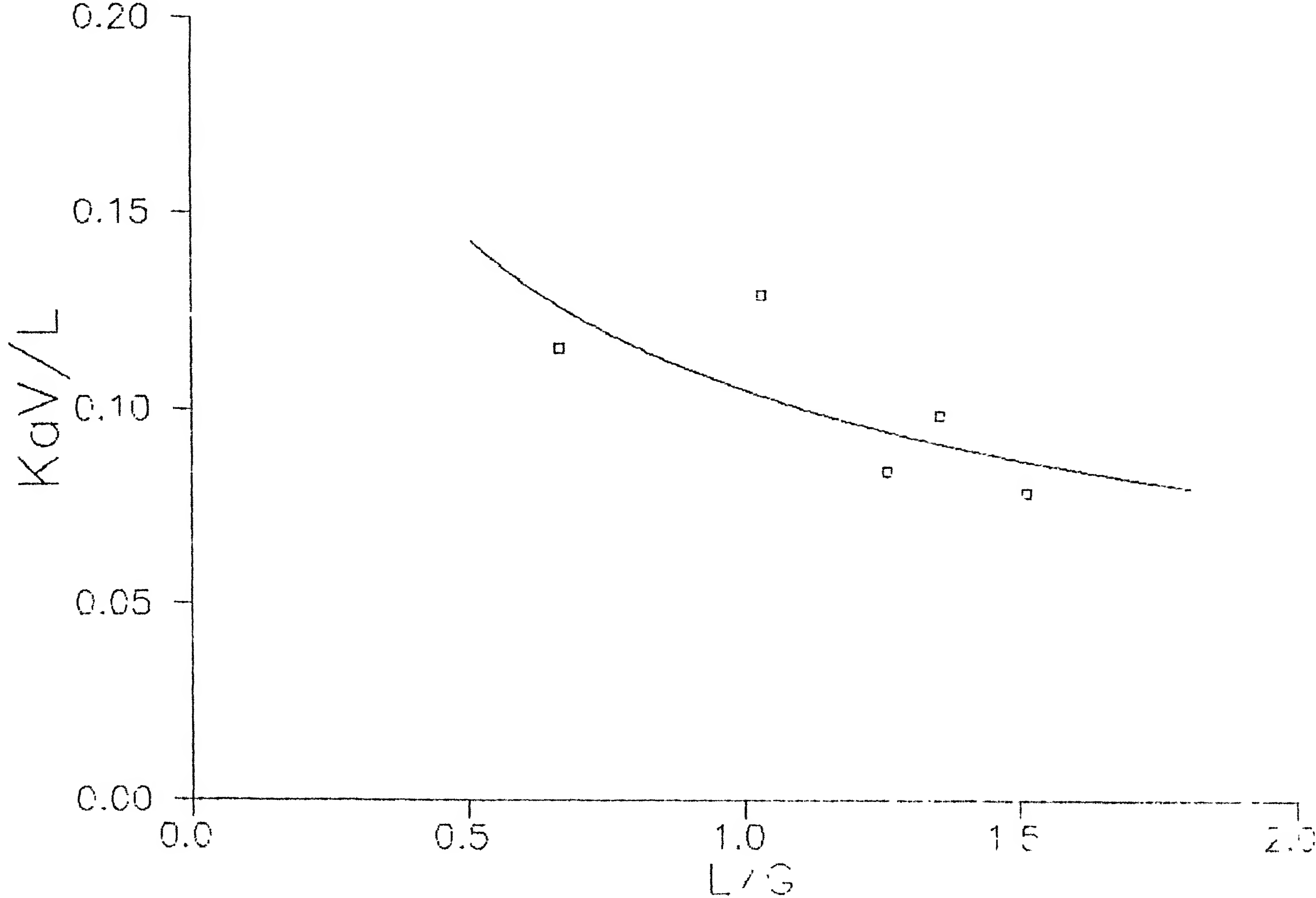


Fig: 4.35
Ka/L vs L/G for the Cellular Packing

found to be 15 to 60 % higher than those of Singh for various angles of orientation. This may be attributed to the increase in cooling of the water due to the presence of packing and a betterwater spray system.

4.5 Packing Characteristics

Film type of packings are commonly used in cooling towers and for a few configurations of these, published information on characteristic curves are available. Usually the data obtained from the packings have been correlated in the following form (Love and Christie, 1961) :

$$Ka/L = \lambda \ (L/G)^{-n} \qquad 4.1$$

This equation gives the volume transfer coefficient which increases with a decrease in the L/G ratio.

The packing used in the present investigation is an unusual design for a packing which is dissimilar to any of the known film type packings. The data recorded on this packing was correlated in similar to the above equation and can be written as:

$$Ka/L = 0.99 \ (L/G)^{-0.45} \qquad 4.2$$

The pressure drop characteristic for this packing is shown in Figure 4.34. In order to assess the effectiveness of this kind of packing more extensive experimentation is necessary for various values of L/G.

Conclusions

1. The drift loss for HDE decreases with increase in the number of stages and decrease in fan speed whereas that for CADE and CDE it also decreases with the increase in the orientation angle of the DE plates.
2. The pressure drop through the DE increases with increase in the number of stages, fan speed and orientation angle.
3. The breakeven values of orientation angle based upon the cost analysis for three stages of CADE and CDE lie between 50° and 60°.
4. The COP of the refrigeration system decreases with increase in the orientation angle.
5. The provision of a packing in the evaporative condenser results in COP values higher than those obtained without the packing.
6. The volume transfer coefficient for a new type of cellular packing is correlated. Pressure drop per unit depth is also determined.

Suggestions for future work

1. Testing of different and newer type of configuration of drift eliminators.
2. Comparative performance evaluation of various designs of drift eliminators.
3. Determination of characteristics by extensive testing of film fill packings, for a range of (L/G) ratios, spray distribution systems etc.
4. Theoretical correlation of the pressure drop across the drift eliminators with the results obtained experimentally.
5. Mathematical modeling for characterizing the plume and drift behaviour from an Evaporative Condenser and a Cooling Tower.
6. Design of devices that can reliably measure fluxes of drift droplets as small as 30 micron diameter and they should be able to do this without perturbing the flow.

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APPENDIX

Method of Calculating KaV/L for the Packing:

The quantity (KaV/L) in the design of cooling towers represents the number of transfer units and is calculated from the Merkel's equation as:

$$KaV/L = \int_{t_{w_2}}^{t_{w_1}} dt_w / (h_{stw} - h) \qquad 1$$

where h_{stw}: is the enthalpy of saturated air at the local water temperature.

h: enthalpy of the air at the corresponding cross-section of the tower.

The integral of equation (1) is solved using Chebycheff's four point method i.e.

$$KaV/L = [(t_{w_1} - t_{w_2})/4] (1/\Delta h_1 + 1/\Delta h_2 + 1/\Delta h_3 + 1/\Delta h_4) \qquad 2$$

t_{w_1}: temperature of water at the inlet to the packing.

t_{w_2}: temperature of water at the outlet of the packing.

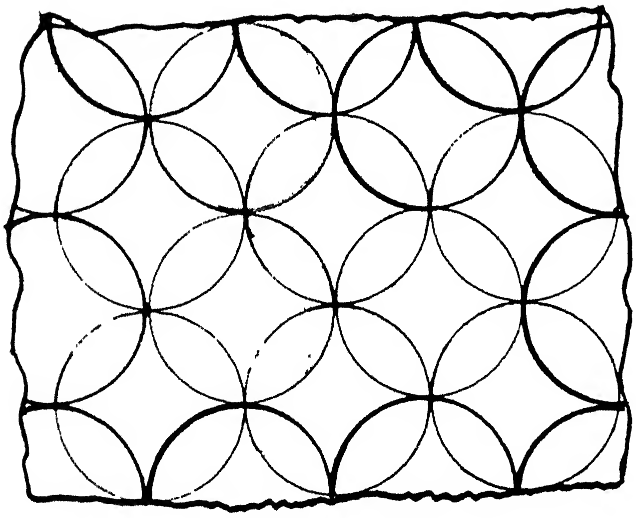
$$\Delta h_1 : [h_{stw} - h] \text{ at } t = t_{w_2} + 0.1 [t_{w_1} - t_{w_2}], \qquad 3$$

$$\Delta h_2 : [h_{stw} - h] \text{ at } t = t_{w_2} + 0.4 [t_{w_1} - t_{w_2}], \qquad 4$$

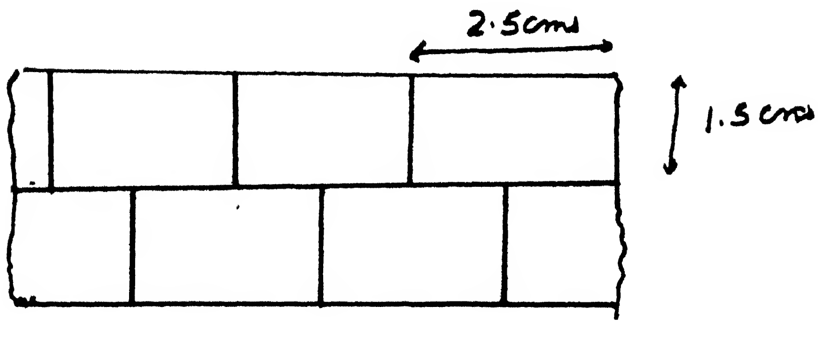
$$\Delta h_3 : [h_{stw} - h] \text{ at } t = t_{w_2} + 0.6 [t_{w_1} - t_{w_2}], \qquad 5$$

$$\Delta h_4 : [h_{stw} - h] \text{ at } t = t_{w_2} + 0.9 [t_{w_1} - t_{w_2}]. \qquad 6$$

The h for equations (3) to (6) are calculated using $h = h_1 + (L/G) (t_{w_1} - t_{w_2})$, where h is the enthalpy of air at the inlet to the cooling tower.



Top View



Side View

Diagram of the Packing & HDE showing two layers.